NASA

FASTENER DESIGN COURSE

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Edited the entire course

A COMMON ERRONEOUS STATEMENT:

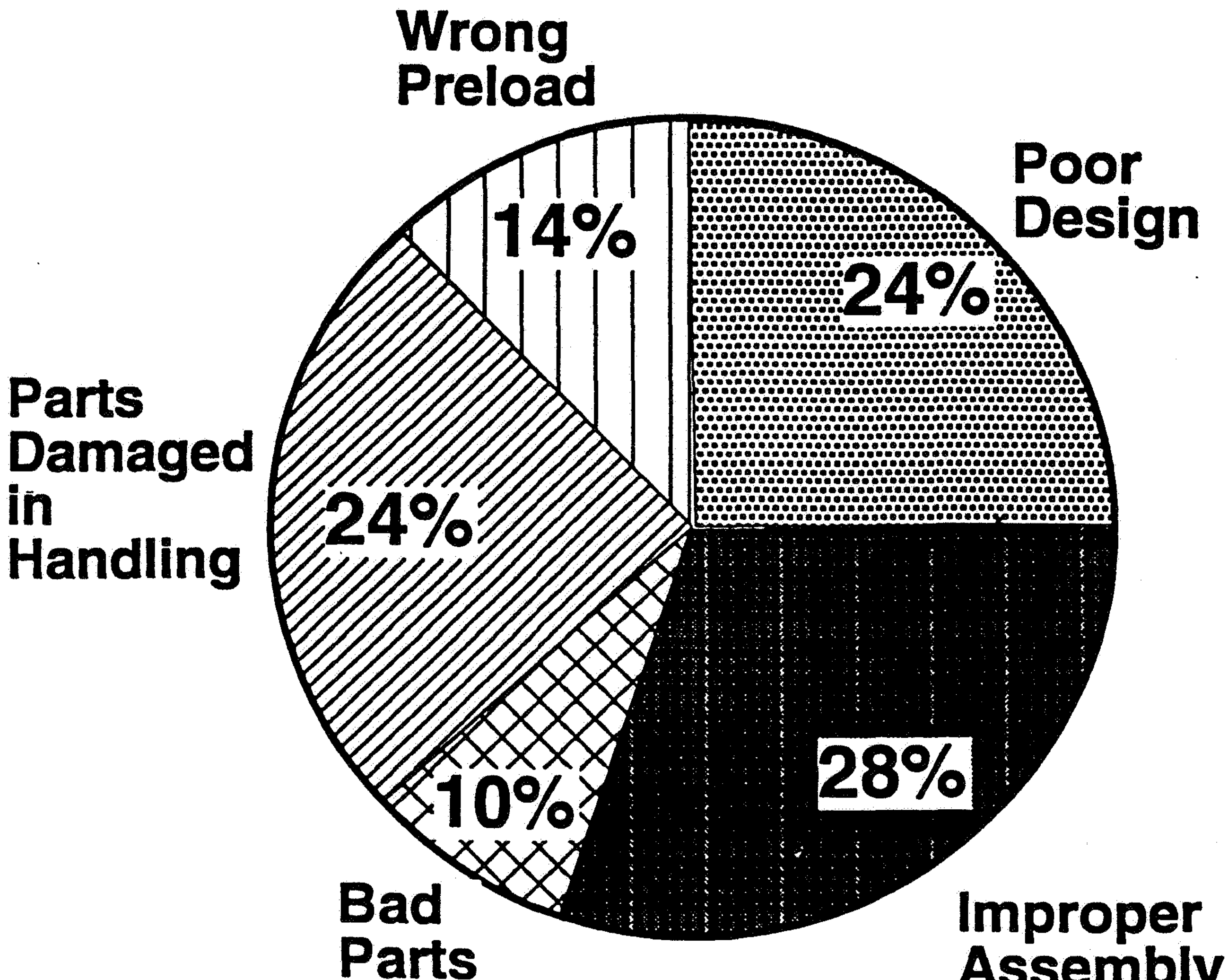
"EVERYBODY KNOWS ABOUT BOLTS AND NUTS"

BUT DO THEY REALLY KNOW?

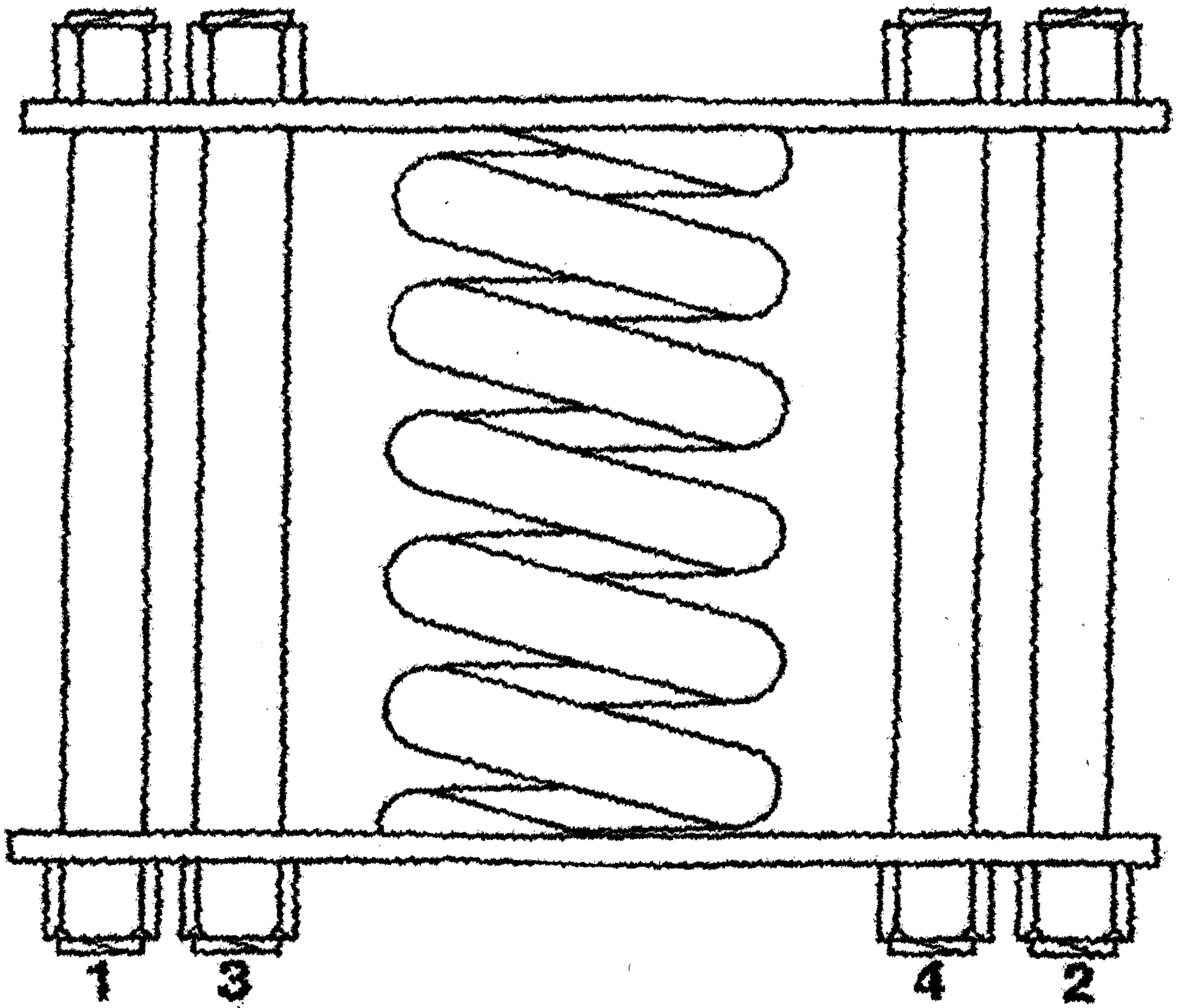
"Seventy-five percent of the assembly labor cost of an automobile is spent on fasteners.

Eighty to eighty-five percent of ALL automobile recalls are fastener related."

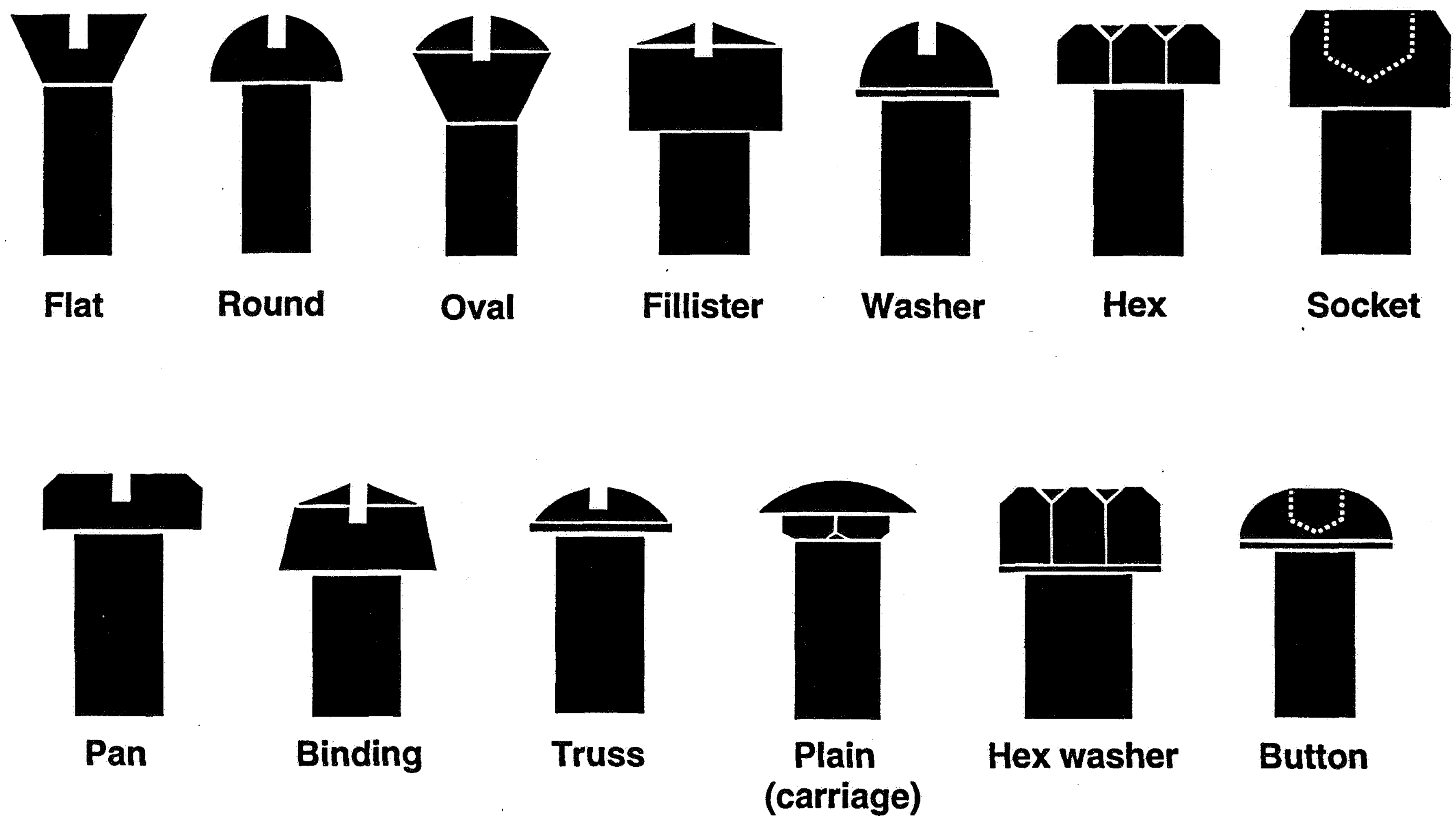
Joseph R. Dudley
V.P., Automotive Marketing
Nylok Fastener Corp.
May, 1995



Causes for Joint Failures in NASA's Sky Lab Program



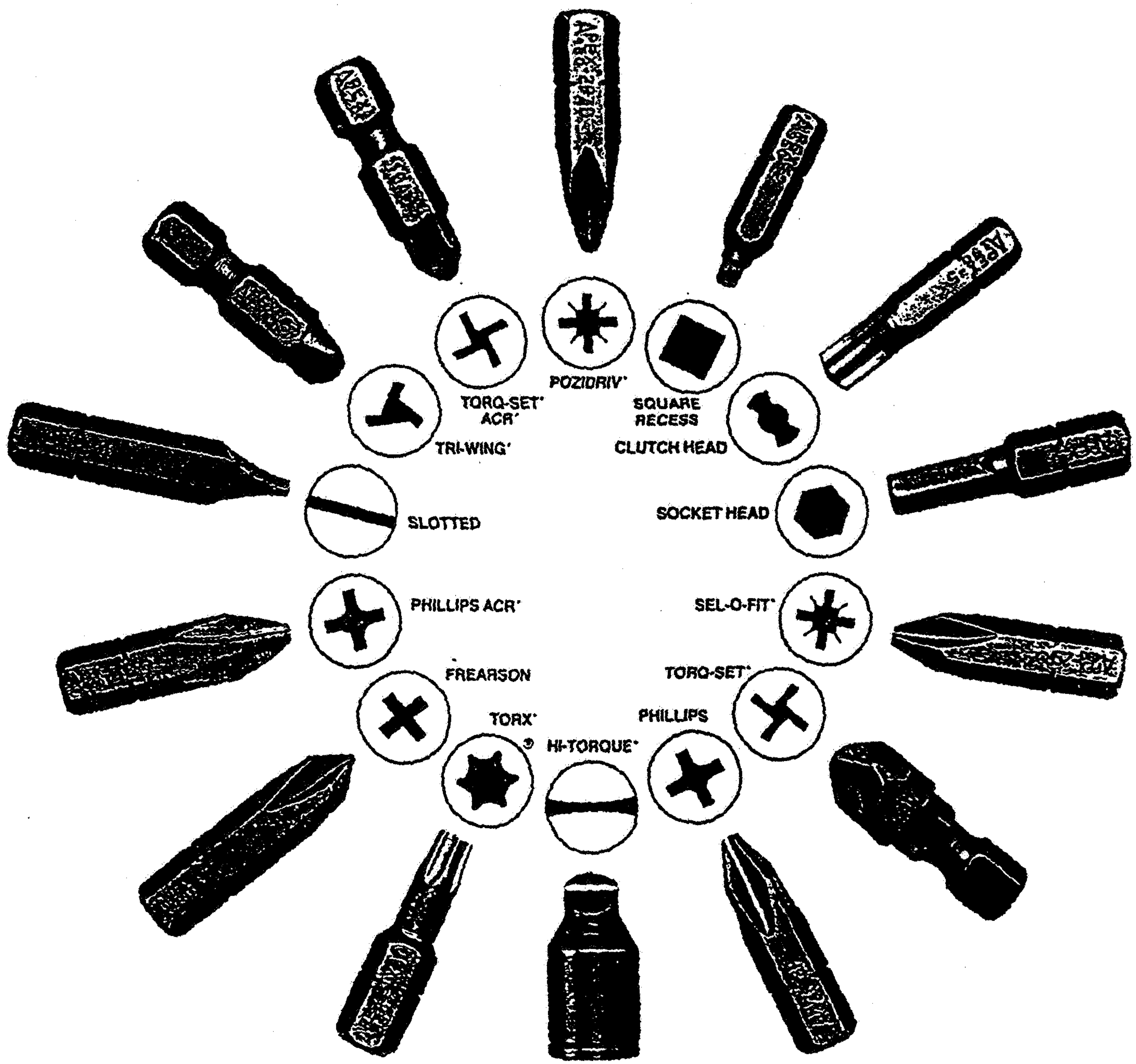
˙olted Joint - Spring Concept



Screw Head Types

APEX®

Internal Drive Systems



AGENDA

- **Topics covered in Fastener Design**

  - Materials
  - Platings and Coatings
  - Thread Lubricants
  - Corrosion
  - Locking Methods
  - Washers
  - Inserts
  - Nutplates
  - Threads
  - Fatigue Resistant Bolts
  - Fastener Torque
  - Joint Stiffness
  - Direct Reading of Fastener Tension
  - Design Criteria
  - Dowel Pins
  - Roll Pins
  - Rivets
  - Lockbolts
  - Inspection and Acceptance of Fasteners
  - Do's and Dont's
  - Frequently Asked Questions
  - Appendices
  - References

FASTENER MATERIALS

- ## Introduction

Fasteners can be made from many materials, but most structural fasteners are made of low carbon steel, alloy steel or stainless steel. Titanium and aluminum bolts have limited usage in the aerospace industry. However, aluminum and titanium rivets are used extensively in the aerospace industry.

Low carbon steel is the cheapest and most common fastener material. It is used for hardware store fasteners which have a typical tensile ultimate of 55 ksi (379 MPa) and is not heat treatable.

Alloy steel fasteners contain enough carbon to be heat treatable and are available up to 300 ksi strength levels.

FASTENER MATERIALS

- **Introduction (Cont'd)**

Carbon steels are not corrosion resistant, so they are normally coated with some type of protective coating.

Stainless steels (including super alloys) are available in a variety of both heat treatable and non-heat treatable alloys with ultimate tensile strengths from 70 ksi to 260 ksi. Note that most of the 400 series materials contain only 12% chromium, which allows some rusting.

A partial summary of fastener materials is given in Tables 1, 2, and 3, and Figures 1, 2, 3, and 4.

FASTENER MATERIALS

- Selection of Materials

  - ➛ Use common fastener materials, strength levels, and coatings whenever possible.

  - ➛ Weight savings versus cost must be evaluated for flight articles.

  - ➛ Galvanic and stress corrosion tolerance levels must be established.

  - ➛ Operating temperatures must be determined before material can be chosen.

  - ➛ The type of loading (static or fatigue) is also a factor in material selection.

FASTENER MATERIALS

- Availability of Materials
  - Common carbon steel fastener materials
    - SAE Grades 1 & 2 (Low Carbon) up to .28 carbon (e.g. 1020)
    - SAE Grade 5 (Medium Carbon) .28 to .55 carbon
    - SAE Grade 8 (Alloy Steel) .28 to .55 carbon but has alloying elements of molybdenum, with chromium and nickel as further additives (e.g. 4037)

BEWARE: SAE J429 allows plain carbon steel to be used for Grade 8 fasteners.

FASTENER MATERIALS

- **Availability of Materials (Cont'd)**
  - Common carbon steel fastener materials
    - ASTM A307 - Equivalent to SAE Grade 1
    - ASTM A449 - Equivalent to SAE Grade 5
    - ASTM A354 - Equivalent to SAE Grade 8
    - ASTM A193 - Contains low carbon and medium carbon steels, such as B5, B6, B7, and B16
    - ASTM A320 - Alloy steels for low temperatures (down to -150 °F)
    - ASTM A325 - Low to medium carbon bolts up to 150 ksi strength
    - ASTM A490 - Low to medium carbon bolts with 150 ksi *MINIMUM* strength (with boron allowed for hardening)

FASTENER MATERIALS

- **Availability of Materials (Cont'd)**

  - Stainless Steel (CRES) and Superalloy Materials

    - 300 series CRES is readily available (up to 80 ksi)

    - A286 CRES is readily available (up to 160 ksi) from aerospace fastener companies in INCH sizes and by special order in METRIC.

    - 400 series CRES is readily available in limited strength levels (up to 125 ksi).

    - Superalloys available only on special order:

- Inconel 718	- MP35N
- 17-7 PH	- MP159
- 17-4 PH	- Inconel X-750
- Waspalloy	- Haynes Alloys
- Titanium	- A286 (above 160 ksi strength)

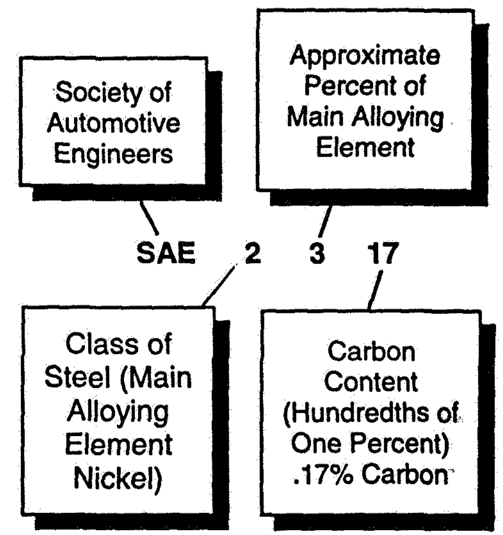
FASTENER MATERIALS

Table 1

AISI-SAE INDEX SYSTEM FOR CARBON AND ALLOY STEELS

Type of Steel	*Series Designation*
Carbon Steels	1XXX
Plain carbon	10XX
Free machining, resulfurized (screw stock)	11XX
Free machining, resulfurized, rephosphorized	12XX
Manganese Steels	13XX
High Manganese Carburizing Steels	15XX
Nickel Steels	2XXX
3.5 percent nickel	23XX
5.0 percent nickel	25XX
Nickel-Chromium Steels	3XXX
1.25 percent nickel, 0.60 percent chromium	31XX
1.75 percent nickel, 1.00 percent chromium	32XX
3.50 percent nickel, 1.50 percent chromium	33XX
Corrosion and heat resisting steels	30XXX
Molybdenum Steels	4XXX
Carbon-molybdenum	40XX
Chromium-molybdenum	41XX
Chromium-molybdenum	43XX
Nickel-molybdenum	45XX and 48XX

Type of Steel	*Series Designation*
Chromium Steels	5XXX
Low chromium	51XX
Medium chromium	52XXX
Corrosion and heat resisting	51XXX
Chromium-Vanadium Steels	6XXX
Chromium 1.0 percent	61XX
Nickel-Chromium-Molybdenum	86XX and 87XX
Manganese-Silicon	92XX
Nickel-Chromium-Molybdenum	93XX
Manganese-Nickel-Chromium-Molybdenum	94XX
Nickel-Chromium-Molybdenum	97XX
Nickel-Chromium-Molybdenum	98XX
Boron (0.0005% boron minimum)	XXBXX



Alloying Elements in Steel

Aluminum	Al	Molybdenum	Mo
Carbon	C	Nickel	Ni
Chromium	Cr	Phosphorus	P
Cobalt	Co	Silicon	Si
Copper	Cu	Tungsten	W
Manganese	Mn	Vanadium	V

Chemistry of

Element	*Steel*	*Cast Iron*
Carbon	.05-1.5%	2.2-3.8%
Phosphorus	.04 Max.	.10-1.00
Sulphur	.05 Max.	.09-.12
Manganese	.30-.90	.40-1.00
Silicon	.15-.30	.50-3.00

Table 2 - Chemical Compositions of Steels

Steels, Alloys and Stainless - Ranges and Limits (Percent)

Grade	C	Mn	P Max.	S Max.	Si Max.	Ni	Cr	Mo Max	Other Elements
Standard Carbon Steels									
1006	.08 max.	.25-.40	0.040	0.050					
1008	.10 max.	.30-.50	0.040	0.050					
1010	.08-.13	.30-.60	0.040	0.050					
1018	.15-.20	.60-.90	0.040	0.050					
1022	.18-.23	.70-1.00	0.040	0.050					
1038	.35-.42	.60-90	0.040	0.050					
1045	.43-.50	.60-90	0.040	0.050					
Rephosphorized & Resulphurized Carbon Steels									
1117	.14-.20	1.00-1.30	0.040	.08-.13					
1141	.37-.45	1.35-1.65	0.040	.08-.13					
1144	.40-.48	1.35-1.65	0.040	.24-.33					
12L14	.15 max.	.85-1.15	.04-.09	.26-.35					Pb (.15-.35)
1215	.09 max.	.75-1.05	.04-.09	.26-.35					
Standard Alloy Steels									
4037	.35-.40	.70-.90	0.035	0.040	.20-.35			.20-.30	
8620	.18-.23	.70-.90	0.035	0.040	.20-.35	.40-.70	.40-.60	.15-.25	
8630	.28-.33	.70-.90	0.035	0.040	.20-.35	.40-.70	.40-.60	.15-.25	
4130	.28-.33	.40-.60	0.035	0.040	.20-.35		.80-1.10	.15-.25	
4340	.38-.43	.75-1.00	0.035	0.040	.20-.35		.80-1.10	.15-.25	
Stainless Steel (Austenitic-Non Magnetic)									
302H	.10 max.	2.00 max.	0.045	0.030	1.00	8.00-10.00	17.00-19.00		
302	.15 max.	2.00 max.	0.045	0.030	1.00	8.00-10.00	17.00-19.00		
303	.15 max.	2.00 max	0.200	.15 min.	1.00	8.00-10.00	17.00-19.00	0.60	
304	.08 max.	2.00 max.	0.045	0.030	1.00	8.00-10.50	18.00-20.00		
316	.08 max.	2.00 max.	0.045	0.030	1.00	10.00-14.00	16.00-18.00	2.00-3.00	
Stainless Steel (Martinsitic - Magnetic)									
410	0.15	1.00	0.04	0.03	1.00		11.50-13.50		
420	.15 min.	1.00	0.04	0.03	1.00		12.00-14.00		
Stainless Steel (Precipitation Hardening Alloy)									
17-4PH	0.07	1.00	0.04	0.03	1.00	3.00-5.00	15.50-17.50		Co+Ta (.15-.45)

FASTENER MATERIALS

- Operating Temperatures

  - -65 °F & Below
    No Carbon Steels
    Only Some Stainless Steels
    Aluminum

  - -65 °F to 450 °F
    Carbon Steels Acceptable
    Stainless Steels
    Corrosion Protection (Usually required for carbon steels)

    - Zinc
    - Cadmium
    - Phosphate

FASTENER MATERIALS

- Operating Temperatures (Cont'd)

  ➔ 450 °F and Above
  Unplated Carbon Steel is Acceptable (up to ~ 700 °F)

  - Silver Plating
  - Nickel Plating
  - Chromium Plating (if heavy coating used)
  - Black Oxide Coating
  - Diffused Nickel Cadmium

  Stainless Steels and Superalloys

TABLE 3 - SUMMARY OF FASTENER MATERIALS

Material	Surface Treatment	Useful Design Temperature Limit, °F	Ultimate Tensile Strength at Room Temperature, ksi	Comments
Carbon Steel	Zinc Plate	-65 to 250	55 and up	--------------
Alloy Steels	Cadmium plate, nickel plate, zinc plate, or chromium plate	-65 to limiting temperature of plating	Up to 300	Some can be used at 900°F
A-286 Stainless	Passivated per MIL-S-5002	-423 to 1200	Up to 220	--------------
17-4PH Stainless	None	-300 to 600	Up to 220	--------------
17-7PH Stainless	Passivated	-200 to 600	Up to 220	--------------
300 Series Stainless	Furnace Oxidized	-423 to 800	70 to 140	Oxidation reduces galling
410,416, and 430 Stainless	Passivated	-250 to 1200	Up to 180	47 ksi at 1200 °F, will corrode slightly
MP35N	None	-423 to 700	260	--------------
MP159	None	-320 to 1100	260	200 ksi at 1100 °F
Inconel 718 Stainless	Passivated per QQ-P-35 or Cadmium Plated	-423 to 900 or Cadmium Plate Limit	Up to 220	--------------
Inconel X-750 Stainless	None	-320 to 1200	Up to 180	136 ksi at 1200 °F
Waspalloy Stainless	None	-423 to 1600	150	--------------
Titanium	None	-350 to 800	160	--------------
Haynes 230	None	to 1800	120	32 ksi @ 1800 °F

FASTENER MATERIALS

- ## Galvanic Corrosion and Stress Corrosion

  ➛ Galvanic Corrosion

  Galvanic Corrosion is set up when two dissimilar materials are in the presence of an electrolyte, such as moisture. A galvanic cell is created by the materials and the most active one (anode) is eroded and deposited on the least active (cathode). Note that the farther apart two materials are on the galvanic corrosion table, the greater the galvanic action between them. The chemicals in the electrolyte can also accelerate the reaction between the two materials.

FASTENER MATERIALS

- **Galvanic Corrosion and Stress Corrosion (Cont'd)**

  - **Galvanic Corrosion (Cont'd)**

    Note that cadmium and zinc are adjacent to aluminum in the galvanic table, which makes them compatible as coatings for steel fasteners used in aluminum.

    To further protect mating surfaces from galvanic corrosion, it is customary to use zinc chromate paste or MIL-S-8802 sealer in fastener holes for aerospace structures. (See Table 8 for the galvanic series.)

TABLE 8 - GALVANIC SERIES FOR SOME COMMON ALLOYS & METALS

(In order of decreasing activity)

Magnesium................(Most Active)
Magnesium Alloys.....
Zinc...........................
Aluminum 5056.........
Aluminum 5052.........
Aluminum 1100.........
Cadmium...................
Aluminum 2024.........
Aluminum 7075.........
Mild Steel..................
Cast Iron....................
Ni-Resist....................
Type 410 Stainless (Active)
Type 304 Stainless (Active)
Type 316 Stainless (Active)
Lead...........................
Tin............................
Muntz Metal..............
Nickel (Active)..........
Inconel (Active)...........
Yellow Brass.............
Admiralty Brass.........
Aluminum Brass........
Red Brass..................
Copper.......................
Silicon Bronze...........
70-30 Copper-Nickel
Nickel (Passive).........
Inconel (Passive)..........
Titanium....................
Monel........................
Type 304 Stainless (Passive)
Type 316 Stainless (Passive)
Silver........................
Graphite....................
Gold.........................(Least Active)

FASTENER MATERIALS

- ## Galvanic Corrosion and Stress Corrosion (Cont'd)

  ➣ ### Stress Corrosion

  Stress corrosion occurs when a sensitive material is loaded in tension in a corrosive environment. The surface will develop pits or cracks from exposure to the corrodent, thus creating stress risers. These stress risers can cause the component to fail at as little as 20% of its calculated load capacity.

  In general, the higher the strength of an alloy steel (and the lower its ductility), the more susceptible it is to stress corrosion cracking. Many stainless steels, such as A286 and Inconels, are not stress corrosion susceptible. 17-4 PH and 17-7 PH are stress corrosion sensitive.

  The design engineer should always consider stress corrosion and galvanic corrosion before selecting his fastener materials.

FASTENER MATERIALS

- ## Decarburization

When medium carbon alloy steels are heat treated to strengths above 180 ksi, carbon can precipitate out of the surface and near surface areas (decarburization). This surface is now not as strong as the parent material. On a machined part, the outer surface can be ground down after heat treatment to remove the decarburized area. However, this is not practical on a threaded fastener, making this strength range undesirable for these materials.

- ## Temper Brittleness

The tempering range of 400 to 800 °F (for many alloy steels above 190 ksi) leaves the material quite brittle. Therefore, fasteners of materials such as 41XX, 86XX, and 43XX should not be used above 190 ksi strength levels.

FASTENER MATERIALS

- **Carbide Precipitation**

Most 300 series stainless steels should not be used above 800 °F, due to carbide precipitation. The carbon will combine with the chromium to form chromium carbide, which is not corrosion resistant. If the chromium content drops below 12%, the steel will rust. This problem can be partially avoided by using 3XXL (low carbon) material and avoided altogether by using 321 (titanium stabilized) or 347 (columbium stabilized) material. The titanium and columbium have a greater affinity for carbon than chromium, so the chromium will stay in solution.

FASTENER MATERIALS

- Material Strengths

After the temperature and corrosion requirements have been determined, the required fastener material strength(s) should be calculated. Keep in mind that the higher the strength of the material, the more stringent the manufacturing and quality requirements become (with increasing costs).

If weight is not critical, it is better to use more fasteners of lesser strength (up to 160 ksi) than to use fewer high strength fasteners.

FASTENER MATERIALS

- Metric Fastener (Property) Classes

The normal way of specifying metric fastener strength is to give a strength "property class" which is the tensile ultimate and yield (as a percent of ultimate) in mega-pascals (MPa). The material *IS NOT SPECIFIED* in the call-out.

An example call-out is "property class 6.8", which is *a carbon steel* with an ultimate strength of 600 MPa and a yield of 0.8 X 600 = 480 MPa. (1 MPa = 145.04 psi).

For *SOME* stainless steels, the property class call-out has a different designation. For 300 series and 400 series stainless, the property class designations are further designated as shown in Figure 1 and Table 4.

FASTENER MATERIALS

- ## Metric Fastener Materials

Unpublicized Fact:

Metric aerospace fasteners are not available in the European market.

- Metric aerospace fasteners are available (usually by special order) from U.S. aerospace fastener manufacturers. They are covered by SAE/AIA specifications NAXXXX and MAXXXX. These specifications are very similar to standard NAS and MS drawings, so they contain dimensions, material specifications, and coatings.
- Metric fasteners of aerospace materials can also be ordered to ANSI specifications.

FASTENER MATERIALS

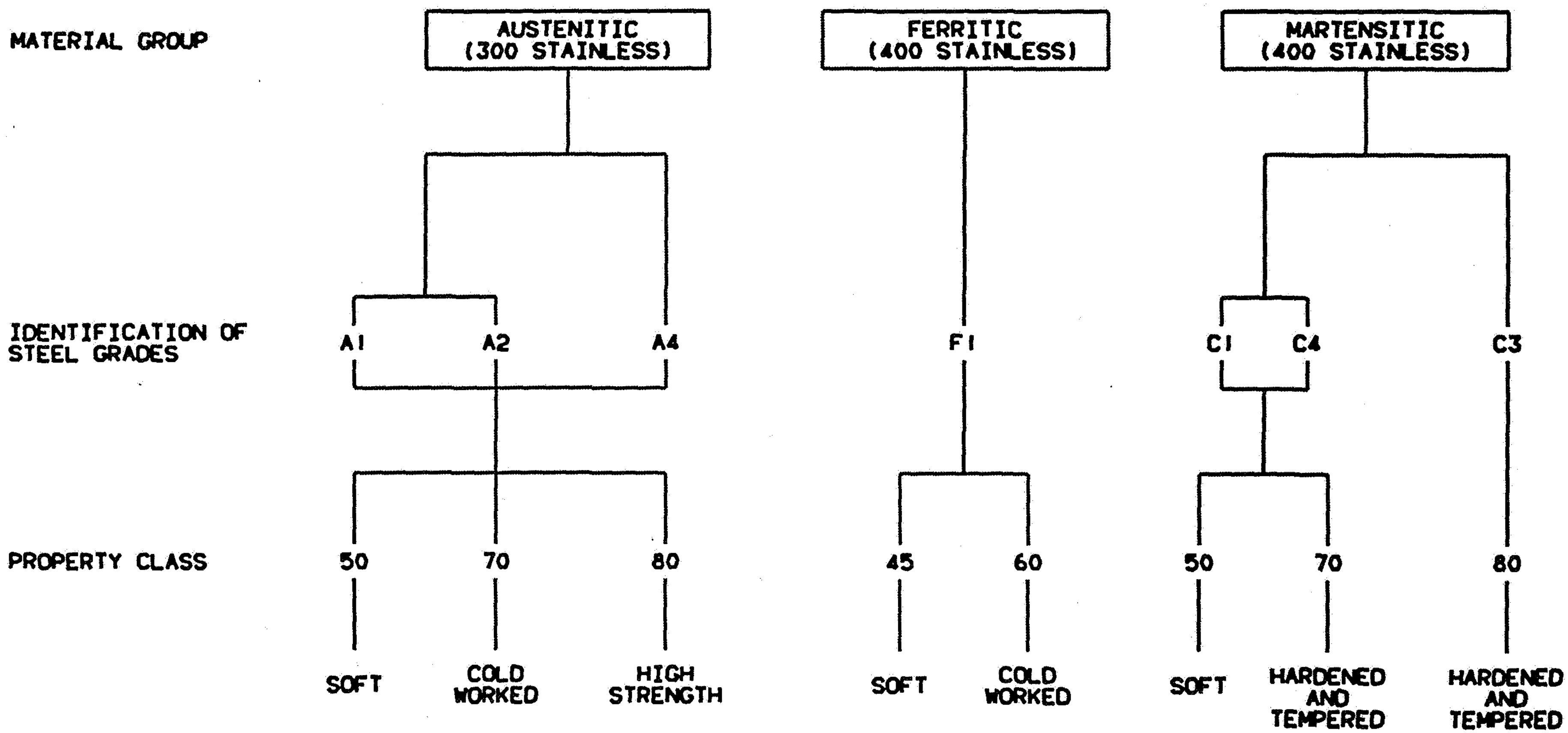


Figure 1 - Metric Fastener Material Classes

FASTENER MATERIALS

Table 4
Metric Stainless Steel Fastener Grades

GRADE	TYPICAL ALLOYS
A1	303S, 303Se
A2	304, 304L, 321, 347
A4	316, 316L, (317, 317L)
C1	410, 420
C3	431
C4	416, 416 Se
F1	430

Figure 2
Family relationships for standard austenitic stainless steels
(Ref. ASM Metals Handbook, Vol. 3, 9th Ed.)

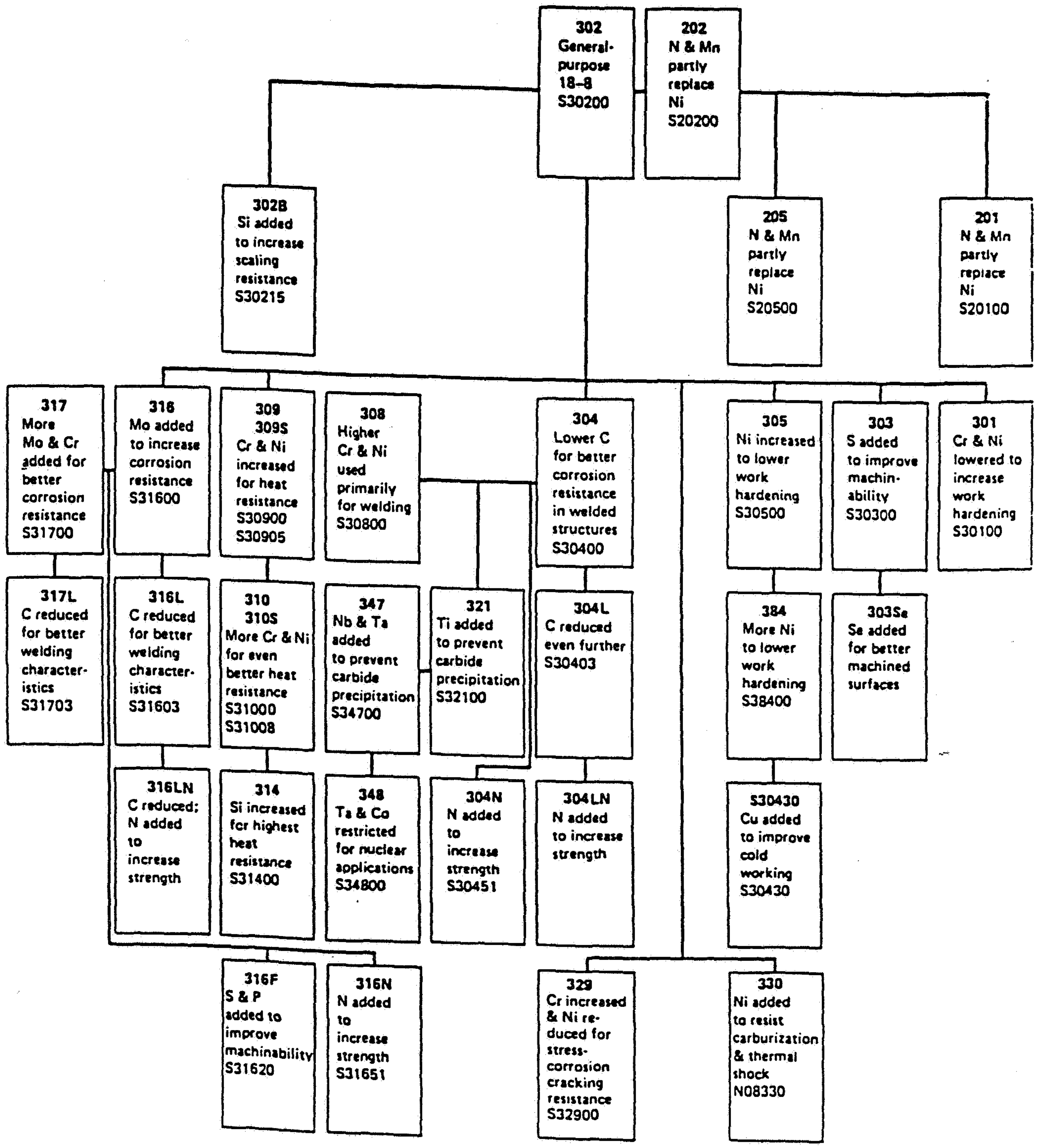


Figure 3
Family relationships for standard martensitic stainless steels
(Ref. ASM Metals Handbook, Vol. 3, 9th Ed.)

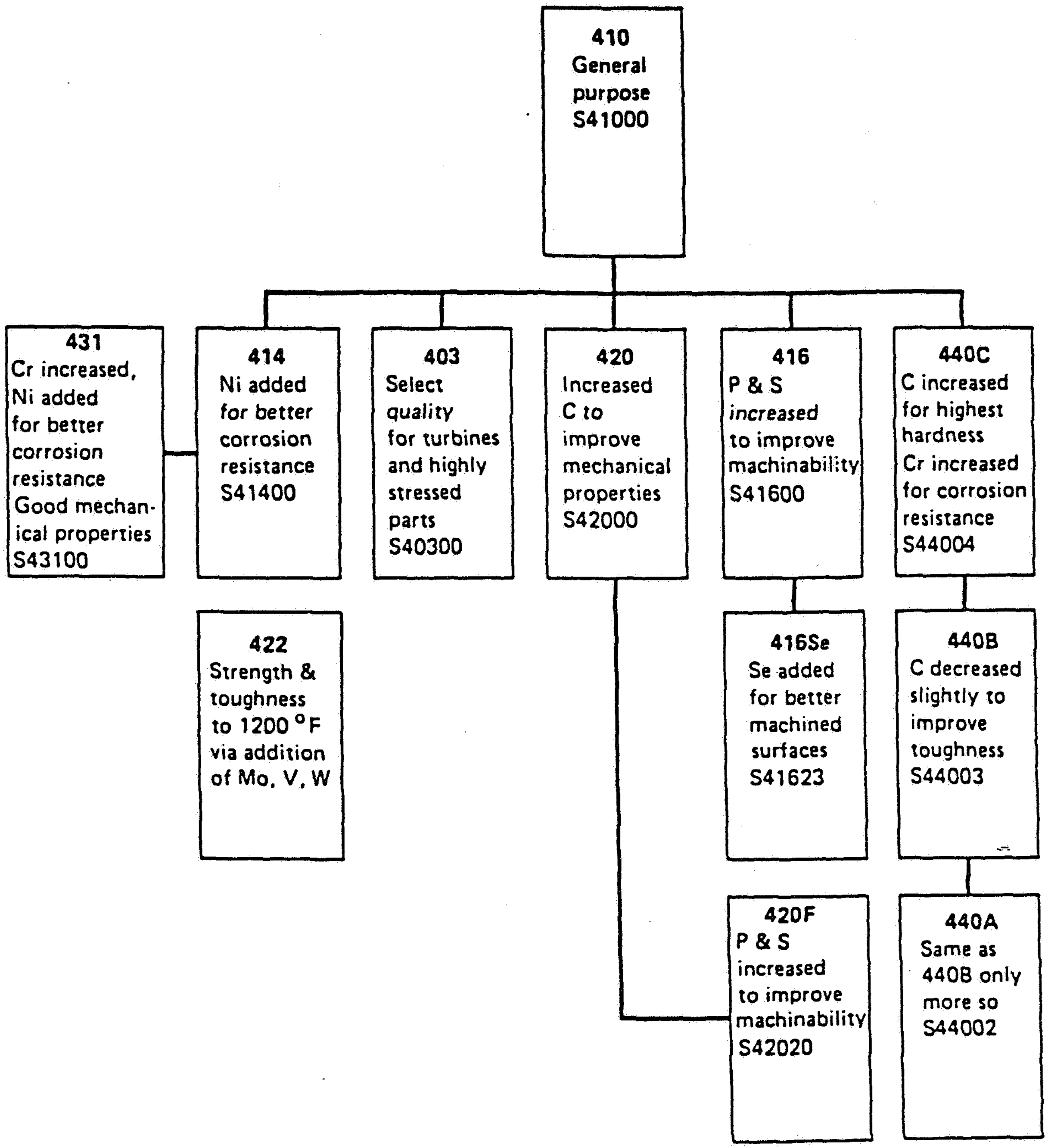
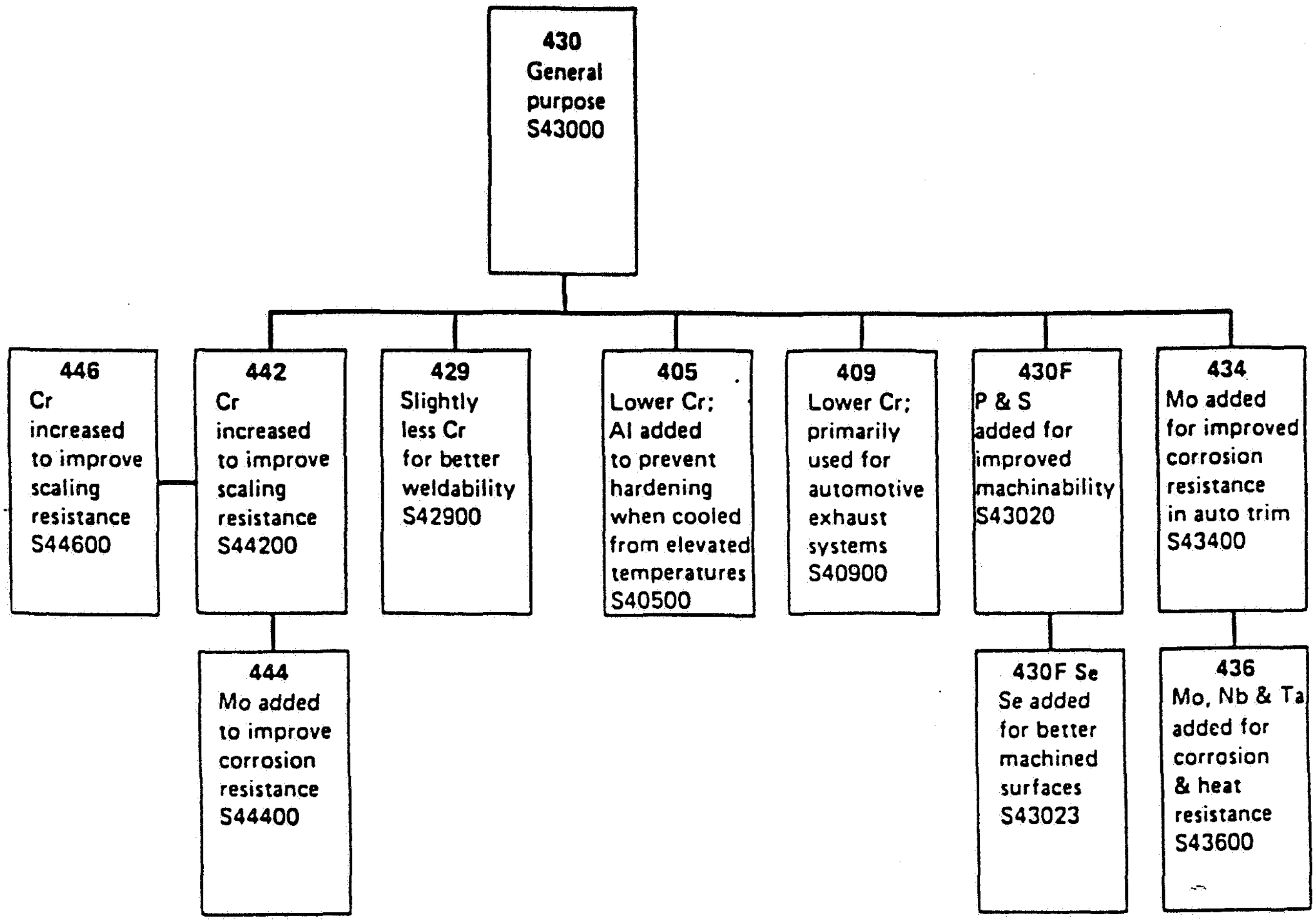


Figure 4
Family relationships for standard ferritic stainless steels
(Ref. ASM Metals Handbook, Vol. 3, 9th Ed.)



GLOSSARY OF TERMS

Fastener Steels:

Alloy Steels Steels alloyed with molybdenum, nickel and chromium (AISI 4037, 4130, 8630) are best where high strength is required. These steels have good cold-forming properties in the annealed condition. They can be heat treated for the best combination of strength, toughness and shock resistance.

Aluminum Alloys Aluminum has good cold-forming characteristics. Many aluminums can be used (2024, 3003, 5056, 5554, 6061, 7075); and where secondary machining is required (2011). Aluminum is corrosion resistant and some alloys can be heat treated.

Carbon Steels Fine grain, fully-killed basic steel with no alloying agent.

> ***Low carbon steel*** range from .06-.18% carbon content (AISI 1006-1018) and have good ductility for cold-forming.
>
> ***Medium carbon steel*** have a .18-.50% carbon content (AISI 1018, 1038, 1041). While stronger and less ductile, these steels respond well to quench and temper.
>
> ***High carbon steels*** [.50% carbon and up (AISI 1066, 1095.)] These are difficult to cold-form unless annealed, but they have high strength and can be heat treated.

Hard-Drawn MB Spring Wire, ASTM A227 Carbon range .60-.70%. Used for general purpose low-cost springs. Commonly available in diameters .031 to .500. It has a lower tensile strength than music wire.

Leaded Steels Lead added to steel improves machinability. Identified by an "L" (AISI 12L14) in the AISI/SAE designation, this is most commonly added to 1100 and 1200 series "screw machine" steels. Leaded steels are not suitable for heat treatment or welding.

Music Wire, ASTM A228 [Carbon range .80-.95%.] High tensile strength can withstand repeated loading. Widely used in small diameter rounds (.005 to .125).

Resulphurized and Rephosphorized Steels (AISI 1117 and 1215) have improved machinability over basic carbon steels. Used commonly for screw machine parts. They are more brittle, less ductile and stronger than equivalent carbon content basic steel. Sulphur acts as an internal lubricant and is the major alloying agent for "1100" series steels. Phosphorous makes the steel more brittle, reducing friction, heat and tool wear. "1200" series steels have both phosphorous and sulphur as alloys.

GLOSSARY OF TERMS

Fastener Steels: (Cont'd)

Stainless Steels

Austenitic stainless steels (AISI 202, 302, 304, 316) are generally non-magnetic and are corrosion resistant due to their large content of nickel chromium. Although they have good cold-forming characteristics, these steels are subject to work hardening and are not heat treatable.

Martensitic stainless steel is a straight chromium steel with little or no nickel (AISI 410, 420). These steels are magnetic and can be heat treated. Higher strength than austenitic types, lower corrosion resistance and harder to cold-form.

Precipitation hardening stainless (AISI 17-4PH) is a nickel chromium stainless with cobalt and tantalum. The alloying agents produce high strength and ductility, good machinability and weldability. Hardening is by aging the steel at 900° for 4 hours followed by air cooling. Not used for cold-forming but commonly used for high-strength, corrosion-resistant screw machine parts.

GLOSSARY OF TERMS

Mechanical Definitions:

Cold Working Deformation of a metal at room temperature without fracture which changes its shape and produces higher tensile strength and machinability.

Ductility The ability of a metal to be deformed extensively under tension load without rupture or fracture. Ductility is expressed in terms of percent elongation and percent reduction of area (e.g., drawn into wire).

Machinability The condition or property of a metal which allows it to be cut, turned, broached or formed by machine tools.

Malleability The ability of a metal to be deformed permanently under compression load without rupture or fracture (e.g., hammer or rolled into sheets).

Tensile Strength The maximum load in tension (pulling apart or shearing) which a material can withstand before breaking or fracturing. Also known as ultimate tensile strength (UTS) or maximum strength.

Work Hardening Hardening that takes place through grain alignment when a metal is bent, rolled or hammered at room temperature. Not all metals work harden.

Yield Strength The load at which a material exhibits a 0.2% permanent deformation. Deformation to determine yield varies with material.

GLOSSARY OF TERMS

Process Definitions:

Alloy Steel A carbon steel to which one or more elements are added to add special properties for a specific use.

Billet A cast section of steel 4 to 6 inches square about 20 ft. long from which dimensional shapes of steel are rolled.

Brass An alloy of copper and zinc which can contain small amounts of aluminum, iron, manganese or tin to produce specific properties.

Bronze An alloy of copper and tin or an alloy of copper, aluminum and silicon.

Carbon Steel A steel in which carbon is the only alloying element added to iron.

Killed Steels Molten steel treated with aluminum, silicon or manganese until no more gas is in the metal and it is in a quiet state.

Pickling The removal of oxide scale from metal by dipping it in a diluted acid bath. The chemically clean surface is then ready for cold rolling or wire drawing.

Rockwell Hardness Test A method of determining metal hardness by indenting with a metal ball or diamond cone under a specified load.

GLOSSARY OF TERMS

Heat Treating Definitions

Annealing Heating and slow cooling to remove stresses, make steel softer, refine the structure and change its ductility.

Carburizing Adding carbon to the surface of iron-based alloys by heating the metal below its melting point in contact with carbon rich solids, liquids or gases.

Case Hardening Carburizing a metal surface followed by quenching to fix a hard outer case high in carbon combined with a relatively soft middle or core.

Cyanide Hardening A method of case hardening which brings the metal surface in contact with molten cyanide salt followed by quenching.

Decarburization Removal of carbon from the surface of steel. This can occur through normal oxidizing action or as the result of heat treatment.

Drawing (tempering) Reheating after hardening, held at a specific temperature and then quenched. This reduces hardness and increases toughness.

Hydrogen Embrittlement A condition where the surface finishing of metal (plating) results in a brittle outer case due to immersion in acid. Baking immediately following the plating process removes this brittle surface condition.

Nitriding A hardening process which adds nitrogen to a metal surface through contact with ammonia gas. This produces surface hardness (case) without quenching.

Precipitation Hardening A hardening process where certain metals are held at elevated temperature without quenching (age hardening).

Quenching Rapid cooling of steel by immersion in oil or water to fix its structure in a hardened state.

Spheroidizing (anneal) Any process of heating and cooling steel that produces a rounded or globular form of carbide. This softens the metal, improving ductility.

Stress Relieve A low temperature heat treatment which removes stresses caused by cold working or welding.

Platings and Coatings

- ## Introduction

Since nearly all commercial fasteners are made of plain carbon or alloy carbon steel, some type of protective coating must be used to prevent corrosion. A multitude of coatings, ranging from oil film to gold can be used to retard or prevent corrosion. For non-carbon steel fasteners, coatings are usually required for either environmental or galvanic corrosion prevention. The secret is to find the coating (or plating) that will give the desired protection at the lowest cost.

Fastener coatings must be thin and must not build up in threads during the coating process. The threads must still be in tolerance after plating or coating.

PLATINGS AND COATINGS

- ## Coating Temperature Limitations

The coating or plating on a fastener is more likely to set the critical temperature limit than the parent fastener material. Some coatings, such as cadmium, can accelerate fastener failure as they decompose from heat exposure. Others, such as oil and black oxide, bake off without harming the parent material.

PLATINGS AND COATINGS

- ## Cadmium Plating

Cadmium is normally applied by electrodeposition to alloy steel fasteners up to 190 ksi. (It should be applied by vacuum deposit for higher strength materials to avoid hydrogen embrittlement).

Cadmium plated parts must be baked at 375 °F for 8 to 23 hours, *WITHIN 2 HOURS AFTER PLATING*, to avoid hydrogen embrittlement.

Cadmium melts at approx. 610 °F, so its service temperature is limited to 450 °F.

PLATINGS AND COATINGS

- **Cadmium Plating** (Cont'd)

Advantages:

➢ Good salt spray resistance
➢ Consistent torque/friction properties
➢ Good galvanic corrosion location
➢ Does not decrease base material fatigue strength

Disadvantages:

➢ Generates cyanide during plating process
➢ Plating and baking must be closely controlled to avoid hydrogen embrittlement
➢ Causes embrittlement of titanium
➢ Expensive
➢ Must be vacuum deposited on high-strength parts to avoid hydrogen embrittlement

PLATINGS AND COATINGS

- ## Zinc Plating

Zinc is probably the most common fastener coating. It is used for hot dip (galvanizing) coating of non-threaded fasteners, but is usually electrodeposited on threaded fasteners.

Zinc plating does not generate the toxic by-products that cadmium generates and zinc is much cheaper than cadmium.

Advantages:

- Zinc plating will heal itself by migrating over scratched areas
- Good galvanic corrosion location

Disadvantages:

- Not as good as cadmium for corrosion resistance
- Torque/tension friction characteristics are inconsistent
- Useful service temperature limit is 250 °F
- Zinc plating can also cause hydrogen embrittlement

Platings and Coatings

- ## Phosphate Coatings

Steel or iron is phosphate coated by treating the material surface with a diluted phosphoric acid usually by submerging the part in a proprietary bath. A mildly protective layer of crystalline phosphate is formed on the material surface. The three principal types of phosphate coatings are zinc, iron and manganese.

Fasteners are usually coated with zinc or manganese phosphates.

PLATINGS AND COATINGS

- **Phosphate Coatings (Cont'd)**

  Advantages:

  - Low cost
  - Can be coated with oil or wax to increase corrosion resistance
  - Phosphate coating is a good primer for painting
  - No hydrogen embrittlement

  Disadvantages:

  - Not a good corrosion resisting coating
  - Inconsistent torque/tension friction properties
  - Phosphate coatings have a temperature limit of 225 to 400 °F

PLATINGS AND COATINGS

- ## Nickel Plating

Nickel plating, with or without a copper strike, is one of the oldest methods of preventing corrosion and improving the appearance of steel and brass. Nickel will still tarnish, much like silver plating, unless it is chromium plated.

Advantages:

- ➛ Will operate in 1100 °F environment
- ➛ Good corrosion resistance

Disadvantages:

- ➛ More expensive than cadmium or zinc
- ➛ Requires baking after plating to prevent hydrogen embrittlement
- ➛ Poor appearance, due to tarnishing

PLATINGS AND COATINGS

• Chromium Plating

Chromium plating is commonly used for automotive and appliance decorative applications, but it can be used for fasteners. However, a copper strike and nickel plating are required for the surface to assure a good chrome plate.

Advantages:

- Use on super strength fatigue resistant carbon steels for corrosion protection (e.g., aircraft landing gear components)
- Can be used up to 1200 °F
- Good appearance

Disadvantages:

- As expensive as stainless steel
- Requires stringent quality control
- Requires baking to prevent hydrogen embrittlement

PLATINGS AND COATINGS

• Ion-Vapor-Deposited (IVD) Aluminum Plating

Ion-vapor-deposited aluminum plating was developed by McDonnell-Douglas for coating of aircraft parts.

Advantages:

- ➢ No hydrogen embrittlement
- ➢ Insulates to deter galvanic corrosion
- ➢ Can be used up to 925 °F
- ➢ Non-toxic by-products from the plating process

Disadvantages:

- ➢ Expensive - plating must be done in a special vacuum chamber
- ➢ IVD is not as good as cadmium in a salt spray test

PLATINGS AND COATINGS

- ## Diffused Nickel-Cadmium Plating

This plating process was developed by the aerospace industry as a high temperature cadmium plating. A nickel coating is first applied on the substrate, followed by a cadmium plating. The fastener is baked for 1 hour at 645 °F.

Advantage:

➣ 1000 °F temperature vs. 450 °F for plain cadmium plating

Disadvantages:

➣ Very expensive
➣ Requires stringent process control
➣ The nickel plate must cover the fastener at all times to avoid cadmium damage to the fastener material
➣ Not recommended for parts above 200 ksi strength

PLATINGS AND COATINGS

- ## Silver Plating

Silver plating is used both to prevent corrosion and as a solid lubricant for fasteners. It is customary to silver plate the nut, rather than the bolt, in a bolted joint.

Advantages:

- Silver plating can be used up to 1600 °F
- Silver plated stainless steel nuts won't gall on stainless steel bolts

Disadvantages:

- Silver plating is very expensive
- Silver tarnishes under normal atmospheric conditions
- Silver should not be used in direct contact with titanium

PLATINGS AND COATINGS

- ## Passivation and Preoxidation

Stainless steel fasteners are normally passivated or preoxidized during the manufacturing process. Passivation is the formation of a near inert oxide coating by treating the surface with an acid. Preoxidation is done by exposing the uncoated fasteners to approximately 1300 °F temperature in an air furnace and cooling in air.

Advantages:

- Deters galling
- Relatively inexpensive process

Disadvantages:

- Mating parts still need to be lubricated for torque coefficient consistency

PLATINGS AND COATINGS

- ## Black Oxide Coating (with oil)

Black oxide is popular for commercial steel fasteners because it gives a shiny black surface which is enhanced by an oil film.

Advantages:

- Very cheap
- No baking required after plating (If material strength is less than 200 ksi.)

Disadvantages:

- Worthless for corrosion prevention once the oil is gone
- Coating does not adhere well to steel

PLATINGS AND COATINGS

- ## Miscellaneous Platings and Coatings

  - Electroless Nickel

    Hard relatively inexpensive surface treatment with nickel for corrosion protection.

  - Sermetel/Sermaloy

    A proprietary aluminum/inorganic coating from TFX Aerospace Company for corrosion protection of both unthreaded and threaded parts. However, thread cycling will degrade these coatings.

  - Synergistic (by General Magnaplate & Tiodize)

    A combination of surface oxidation and fluropolymers used for both corrosion protection and lubricity.

PLATINGS AND COATINGS

- **Proposed Replacements for Cadmium**
  - Dacromet 320 with PLUS L Sealer (Metal Coating International of Chardon, Ohio)
    - Proprietary coating of metal oxides plus zinc and aluminum
    - PLUS L is a clear sealer used to give better friction properties
    - Good up to 600 °F
    - Coating is damaged by assembly/disassembly cycles
    - Dacromet will support fungus growths (Cadmium won't)
    - Comparative cost figures are about the same for Dacromet and Cadmium
  - Zinc-Nickel Coating
    - This coating is approximately 90% zinc and 10% nickel
    - Zinc-nickel coating is not recommended for fasteners (torque coefficient varies)

TABLE 5 - SUMMARY OF PLATINGS AND COATINGS

Type of Coating	Useful Design Temperature Limit, °F	Remarks
Cadmium	450	Most common for aerospace fasteners
Zinc	140 to 250	Self-healing and cheaper than cadmium
Phosphates: Manganese Zinc Iron	 225 225 to 375 400	Mildly corrosion resistant but main use is for surface treatment prior to painting. Another use is with oil or wax for deterring corrosion.
Chromium	800 to 1200	Too expensive for most applications other than decorative
Silver	1600	Most expensive coating (Also anti-galling)
Black Oxide (and oil)	300a	Ineffective in corrosion prevention
Preoxidation (CRES) fasteners only	1200	Deters freeze-up of CRES threads due to oxidation after installation
Nickel	1100	More expensive than cadmium or zinc
SermaGard and Sermatel W	450 to 1000	Dispersed aluminum particles with chromates in a water-based ceramic base coat
Stalgard	475	Proprietary organic and/or organic-inorganic compound used for corrosion resistance and lubrication (in some cases)
Diffused Nickel-Cadmium	900	Expensive and requires close control to avoid hydrogen damage

a) Boiling Point of Oil

Table 6

FASTENER PLATING AND FINISHES

Finish	Color	For Use On (Material)	Corrosion Resistance	Characteristics
Zinc (electroplated)	White to blue grey	All metals	Good	Most common used plating. Good rust-resisting qualities, appearance and low cost.
Cadmium (electroplated)	Bright or dull silver grey	All metals	Excellent	Superior rust-resisting qualities used in marine and aviation applications. Relatively high cost and toxic to the environment.
Chromate	Yellow, olive drab, black, blue/white	Zinc & cadmium plated parts	Very good	A secondary dipping process after plating increasing corrosion resistance, adding color or brilliance.
Dichromate	Yellow, brown, green or irridescent	Zinc & cadmium plated parts	Very good	Secondary dip same as chromate only with a rainbow appearance.
Black Zinc	Black	All metals	Very Good	Shining black appearance with good rust-resisting qualities.
Black Oxide	Black	Ferrous metals and stainless steel	Fair	A chemical discoloration which does not add to part thickness. Usually combined with an oil dip. Rust resistance comes from the oil only.
Phosphate & Oil	Charcoal grey or black	Steel	Good	Zinc or manganese phosphate used with a rust-inhibiting oil dip. Low cost.
Color Phosphate	Blue, green, red, purple, etc.	Steel	Very Good	Chemically produced coating superior to regular phosphate and oil.
Iridite	Olive drab, green, black, red, blue, bronze	All metals	Good	Applied on top of zinc or cadmium plating as a die for color and additional corrosion protection.
Nickel	Silver	All metals	Very good	Hard stable finish, relatively expensive and sometimes hard to apply.
Chromium	Bright blue/white	All metals	Very good	Hard lustrous finish adds wear resistance and is very expensive.
Hot Dip Zinc (galvanizing)	Dull grey	All metals	Very Good	Parts are dipped in pure zinc. Gives maximum corrosion protection. Adds a thick irregular coating. Size must be adjusted to allow for thickness of coat.
Passivating	Bright - etched	Stainless steel	Excellent	Parts are dipped in nitric acid which removes iron particles and brightens the finish. Produces a passive corrosion-resistant finish.
Anodizing	Frosty - etched	Aluminum	Excellent	Acid dip produces a hard oxide surface. Can be color dipped after anodizing for preferred finish.

THREAD LUBRICANTS

- Introduction

There are many thread lubricants available. The most common ones are oil, grease or wax, graphite, silver and molybdenum disulfide. There are also several proprietary lubricants such as Never-Seez, Silver Goop, Synergistic, and Everlube coatings. Some coatings are applied at installation and some are cured on the fastener by the manufacturer.

THREAD LUBRICANTS

- Oil and Grease
  - Good lubrication up to the boiling point of the oil or grease (approx. 250 °F)
  - Can't be used in vacuum
- Graphite
  - "Dry" graphite is really not dry. It is a fine carbon powder that needs moisture (oil or water) to become a lubricant. When the moisture evaporates it becomes an abrasive powder.
  - Can't be used in vacuum

THREAD LUBRICANTS

- Silver Plating
  - ➣ Normally used on CRES nuts with CRES bolts as both a lubricant and anti-galling coating
  - ➣ Can be used up to 1600 °F
  - ➣ Can be used in a vacuum
  - ➣ Very expensive
- Molybdenum Disulfide
  - ➣ Most popular dry lubricant
  - ➣ Temperature limit of 750 °F
  - ➣ Can be used in a vacuum
  - ➣ Can be applied to both steel and CRES

THREAD LUBRICANTS

- Never-Seez
  - ➛ A proprietary petroleum-base lubricant and anticorrodent, containing metal oxides (usually copper or nickel)
  - ➛ Good up to 2200 °F, since metal flakes remain between threads as base boils off
  - ➛ Must be reapplied for each reassembly
  - ➛ Can't be used in vacuum

THREAD LUBRICANTS

- Silver Goop
  - ➛ A proprietary compound (paste) containing 20 to 30% silver
  - ➛ Can be used up to 1500 °F
  - ➛ Not to be used on aluminum or magnesium
  - ➛ Very expensive
  - ➛ Can't be used in vacuum

THREAD LUBRICANTS

- Fluorocarbon Coatings
  - Several proprietary coatings such as Synergistic, Standcote, Stalgard and Everlube are on the market.
  - These coatings are good for only a few (1 to 3) assembly cycles
  - Can be used up to approx. 400 °F
  - Can be used in a vacuum
- Milk-of-Magnesia (Magnesium Hydroxide)
  - Used by some turbine engine companies for engine assembly

TABLE 7 - SUMMARY OF THREAD LUBRICANTS

Type of lubricant	Useful design temperature limit, °F	Remarks
Oil or grease	250	Most common; cannot be used in vacuum
Graphite	up to 250°	Cannot be used in vacuum
Molybdenum disulfide	750	Can be used in vacuum
Synergistic Coatings	500	Can be used in vacuum
Silver Plating	1600°	Don't use on aluminum or magnesium parts
Neverseez	2200	Because oil boils off, must be applied after each high-temperature application
Silver Goop	1500	Do not use on aluminum or magnesium parts; extremely expensive
Everlube	500	Gives good torque coefficients but shouldn't have many assembly/disassembly cycles

CORROSION

- **Introduction**

Corrosion of metals is a major field by itself, so only a brief discussion of fastener corrosion will be covered here.

Galvanic corrosion and stress corrosion were covered in the materials section. The corrosion resistance of a particular metal to a corrodent can be found in a book of tables such as *"Corrosion Resistance Tables"*, by Philip Schweitzer, Marcel Dekker Publishing Co. (2 volumes).

Hydrogen embrittlement and graphite corrosion will be covered in this section.

CORROSION

- ## Hydrogen Embrittlement

Hydrogen embrittlement is talked about and tested for, but it is still a somewhat mysterious occurrence. It is caused by having free hydrogen ions in the presence of the metal (normally steel) during manufacturing or plating. The higher the strength of the material, the more sensitive it is to hydrogen embrittlement.

➢ ### Hydrogen Chemical Reaction

Hydrogen can combine with carbon in a steel to form methane gas or with alloying elements such as titanium, niobium (columbium), or tantalum to form hydrides. The methane gas can cause cracks and the hydrides are weaker than the parent material.

CORROSION

- **Hydrogen Embrittlement** (Cont'd)
  - Hydrogen Blistering
    - Atomic hydrogen can diffuse into voids in the material and combine into molecules. The hydrogen molecules can't diffuse back through the material so they build up pressure to create blisters which eventually burst.
    - There is no external indication that hydrogen is present.
  - Hydrogen Environment Embrittlement
    - If a pressure vessel contains hydrogen gas at very high pressures (2000 psi & up), any moisture present can ionize to allow atomic hydrogen to penetrate the vessel wall.
    - It is unlikely that fasteners would be installed inside such a chamber.

CORROSION

- **Hydrogen Embrittlement (Cont'd)**
  - **Hydrogen Embrittlement Prevention**
    - Use killed (fully deoxidized) steels for fasteners
    - Coat/plate the fasteners
    - Bake hydrogen out within two hours after plating
    - Tailor plating bath to minimize free hydrogen ions
    - Avoid the use of alloy steel fasteners above 190 ksi
    - Use stainless steels, such as A286 and MP35N, which are not susceptible to hydrogen embrittlement
    - Test suspect fasteners for embrittlement prior to acceptance

CORROSION

- **Graphite Corrosion**
  - ➛ Graphite in a resin bonded dry film lubricant which will cause corrosion when exposed to moisture
  - ➛ Dry graphite is an abrasive
  - ➛ Graphite is not recommended for lubrication of locks
- **Dezinctification**
  - ➛ A general term for drastic removal of a particular element from a material by corrosion
  - ➛ Usually means the removal of zinc from brass by chemical action, leaving a brittle shell of copper

TABLE 8 - GALVANIC SERIES FOR SOME COMMON ALLOYS & METALS

(In order of decreasing activity)

Magnesium................(Most Active)
Magnesium Alloys.....
Zinc...........................
Aluminum 5056.........
Aluminum 5052.........
Aluminum 1100.........
Cadmium...................
Aluminum 2024.........
Aluminum 7075.........
Mild Steel..................
Cast Iron....................
Ni-Resist....................
Type 410 Stainless (Active)
Type 304 Stainless (Active)
Type 316 Stainless (Active)
Lead..........................
Tin............................
Muntz Metal...............
Nickel (Active)..........
Inconel (Active)...........
Yellow Brass.............
Admiralty Brass.........
Aluminum Brass........
Red Brass..................
Copper.......................
Silicon Bronze...........
70-30 Copper-Nickel
Nickel (Passive).........
Inconel (Passive)..........
Titanium....................
Monel........................
Type 304 Stainless (Passive)
Type 316 Stainless (Passive)
Silver........................
Graphite....................
Gold.........................(Least Active)

LOCKING METHODS

- ## Introduction

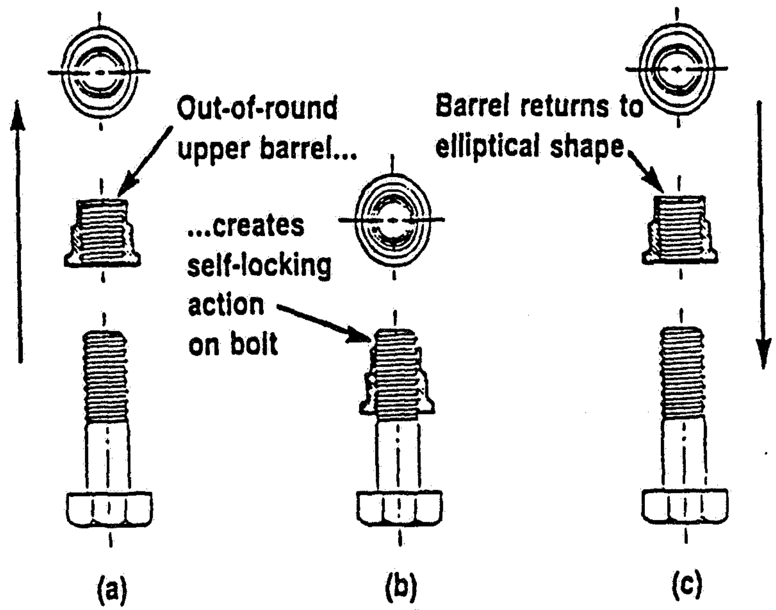
In most applications, some type of locking must be used to prevent the threaded fastener from loosening under load. Without a locking device, the only resistance to loosening is thread friction and head (and/or nut) friction.

The use of fine threads gives slightly better resistance to loosening from vibration than coarse threads, due to the flatter angle on the threads.

It is common practice to mount bolts with heads up to lessen the loss of loose bolts.

Locking Methods

- Deformed Thread

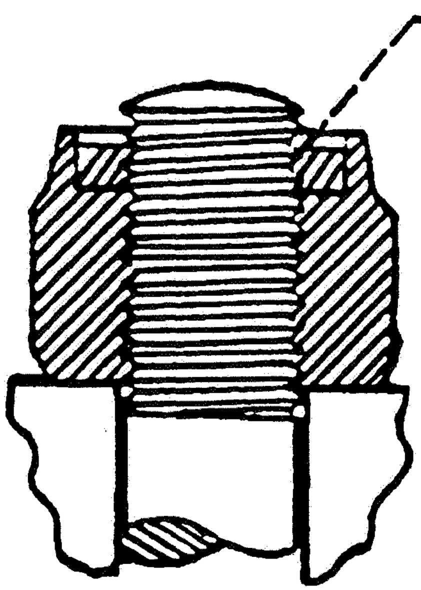


(a) Before assembly
(b) Assembled
(c) After withdrawal

- Nut can be formed in one operation.
- Operating temperature range is limited only by the parent material, its plating, or both.
- Nut can be reused approximately 10 times before losing its locking capability.

LOCKING METHODS

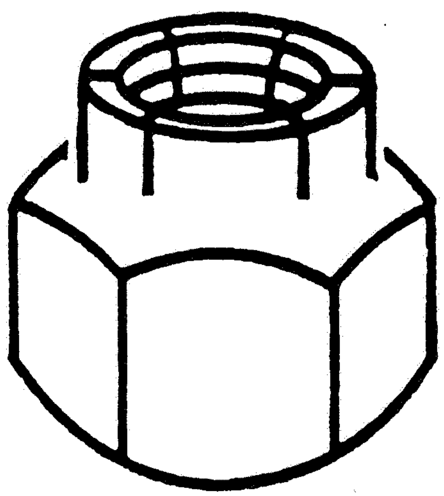
- Locking Collar



- A fiber or nylon collar (washer) is mounted in the top of the nut.
- The collar has a smaller internal diameter than the bolt diameter, so it interferes on the bolt threads, providing locking action.
- The collar provides some sealing from external or internal liquids.
- The temperature limit for the collar is approximately 250 °F.

LOCKING METHODS

- Split-beam Locknut

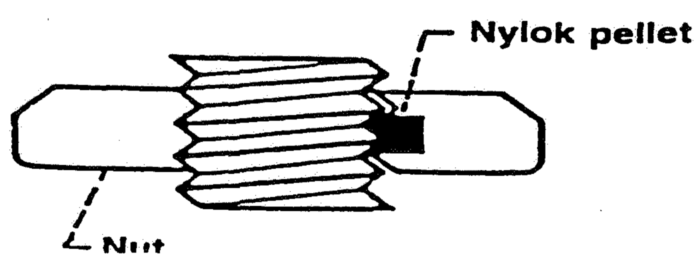


Full-height, heavy-duty hex

- Nut has slots in the top with undersize thread diameter in slotted area.
- Nut spins freely until bolt reaches the slotted area.
- Split "beam" segments deflect outward, creating locking by binding of the mating threads.

LOCKING METHODS

- **Nylok Pellet**
  - ➢ The pellet (usually made of nylon or teflon) is installed in either the nut threads or bolt threads. The pellet is press-fitted into a hole or a slot in the threads and protrudes outside the major diameter of the threads.
  - ➢ Locking action occurs when pellet is compressed by mating threads.
  - ➢ If a nut is used, the pellet will be in the nut.
  - ➢ If a bolt is installed in a tapped hole, the pellet will be in the bolt.
  - ➢ Maximum operating temperature of the pellet is 250 °F.
  - ➢ The pellet damages quickly during assembly/disassembly cycles.



LOCKING METHODS

- **Locking Adhesives**

  ➛ Loctite™ (single adhesive)

  This type of adhesive is made by several manufacturers in different grades to match the locking capabilities to the frequency of fastener removal. For example, Loctite 242 is for removable fasteners and Loctite 271 is for tamper-proof fasteners. Other manufacturers of single adhesive locking compounds are Bostic, ND Industries, Nylok, 3M, Felpro, and Permabond.

  Loctite can be used up to approximately 400 °F.

LOCKING METHODS

- ## Locking Adhesives

  ➢ **Epoxy Ribbon**

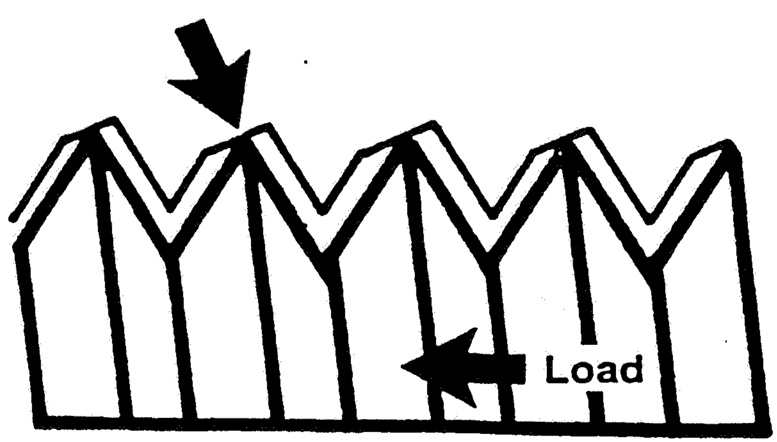
  This adhesive is usually put on as a two layer ribbon by the manufacturer and can be stored. The epoxy is mixed and activated when the bolt or nut is installed.

  These epoxies have a limited amount of thread sealing capability if they are applied in the circumferential direction.

  The maximum operating temperature for epoxy ribbons is also about 400 °F.

LOCKING METHODS

- Spiralock™ Thread



This thread features a special tap which gives wedge ramps on the tapped hole. These ramps elastically deform the crests of the bolt threads to provide friction locking. The load is distributed over several threads, which increases the locking capability. Tests have shown that this type of thread is satisfactory for moderate resistance to vibration.

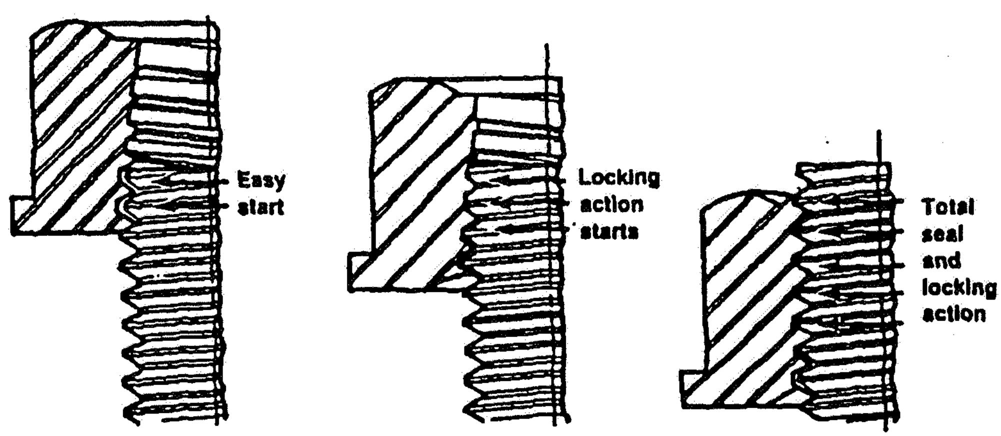
LOCKING METHODS

- **Direct Interfering Thread**

  A direct interfering thread (external) has an over-sized root diameter to give a slight interference fit for locking purposes. Some studs have this feature.

- **Tapered Thread**

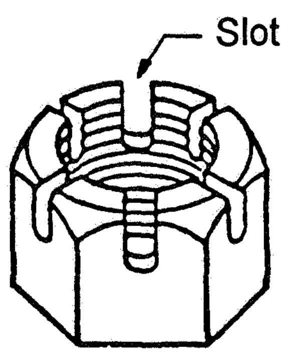
  The tapered thread is a variation of the direct interfering thread, except that the minor diameter is tapered to interfere on the last 3 or 4 threads of a nut or bolt. (See Figure below.)

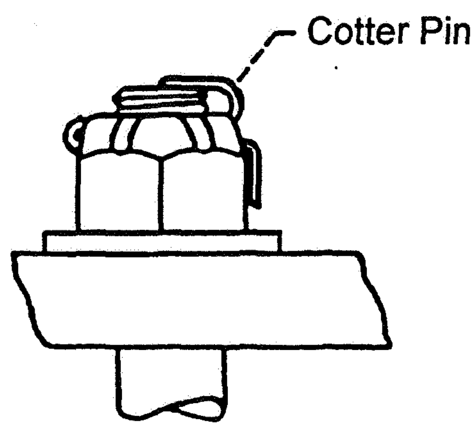


Locking Methods

- **Castellated Nut with Cotter Pin**

Slotted Nut



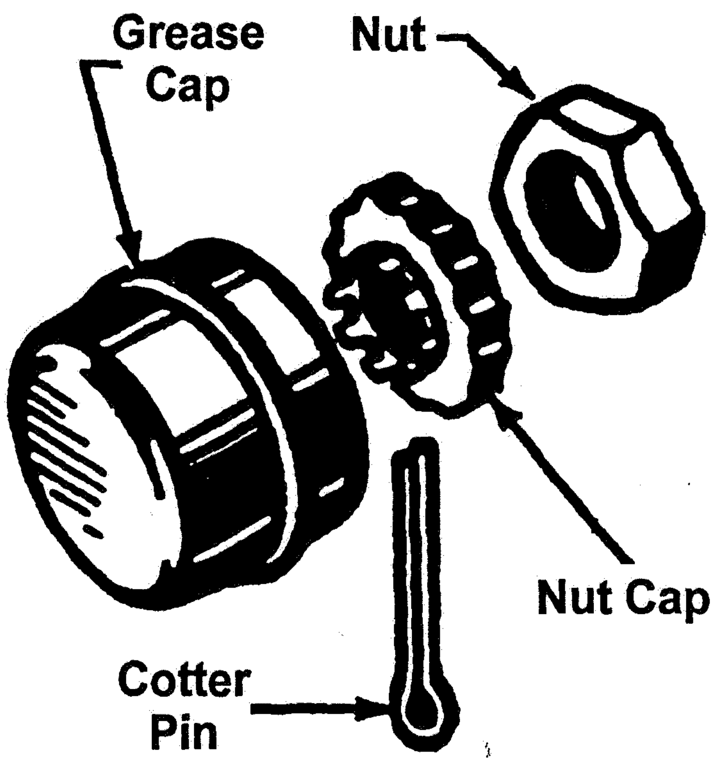


Cotter Pin Locking

The castellated nut normally has six slots as shown above. The mating bolt has a single hole in its threaded end. The nut is torqued to the desired torque value. It is then rotated forward or backward (depending on the user's preference) to the nearest slot that aligns with the drilled hole in the bolt. A cotter pin is then installed as shown above. This nut works extremely well in low torque applications such as holding a wheel bearing in place.

LOCKING METHODS

- Nut Cap and Cotter Pin



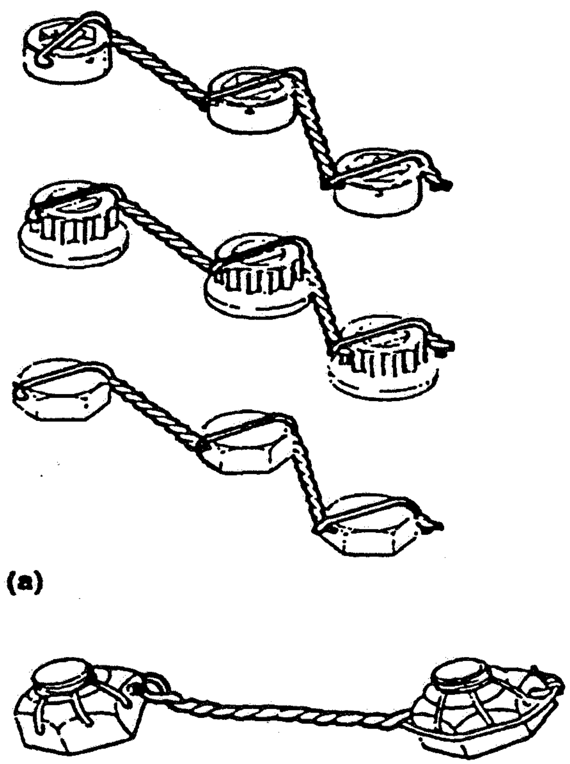
- This locking device is a sheet metal stamping which fits over a standard nut (as shown below) to lock like a castellated nut and cotter pin assembly.

- This cap has slotted tabs on its ID to provide locking with a cotter pin.

- It is widely used on automotive axle/bearing installations.

LOCKING METHODS

- Lockwiring



(a)

(b)

(a) Multiple fastener application (double-twist method, single hole).
(b) Castellated nuts on undrilled studs (double-twist method).

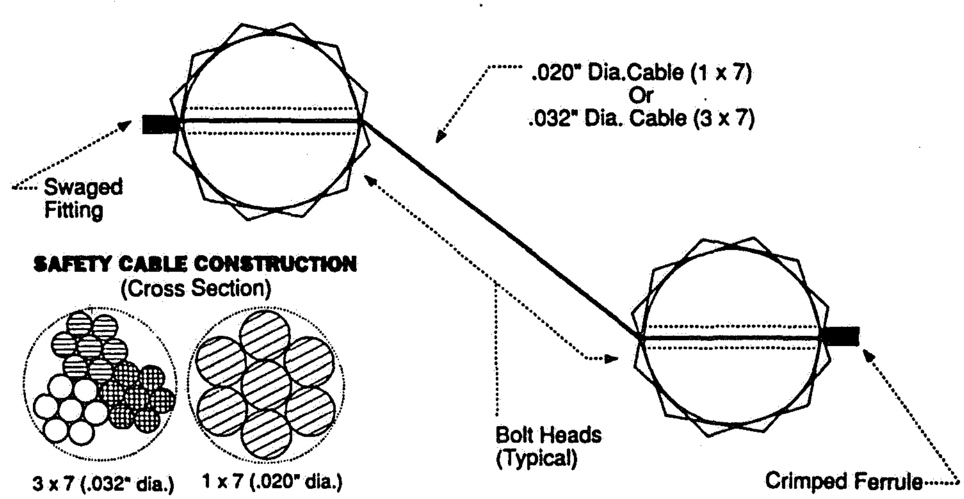
Lockwiring is labor intensive, but it is frequently using for locking a circular pattern of bolts in tapped holes. The bolts usually have through holes in the heads as shown. The nuts are corner drilled. The principle of lockwiring is to tie the heads or nuts together by lacing the wire such that loosening rotation by one fastener must tighten the adjacent ones.

Lockwiring is covered by MS33540.

LOCKING METHODS

- **Bergen Cable Locking (Bergen Cable Technologies Co.)**

This is a faster (and cheaper) method of lockwiring which uses a cable with a swaged connector on the starting end. When the last fastener is reached, the cable is tightened and a ferrule is crimped on the end of the cable to retain the tensile load. The tensioning and crimping are done with a patented tool to avoid human error. (See NASA CR-4473 for more information.)



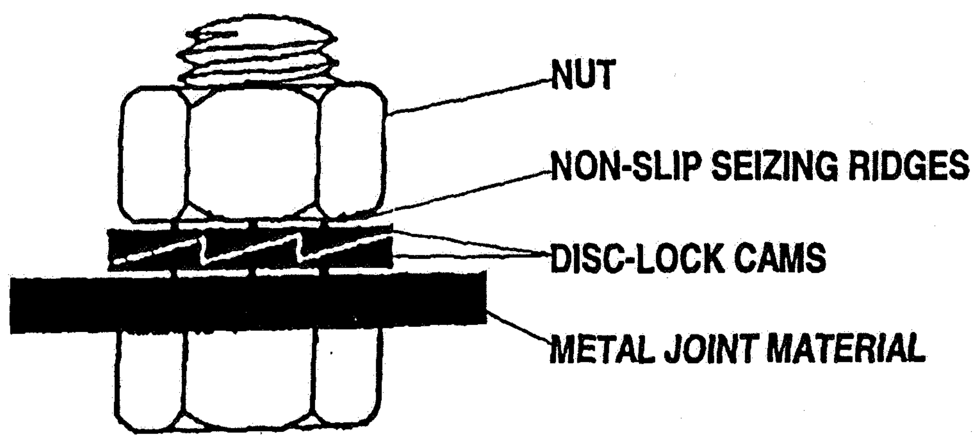
LOCKING METHODS

- ## Disc-Lock Washers

These washers are used in pairs as shown in the figure below. The bottom of each washer has ridges which must embed in the joint material and the nut/bolt head to prevent washer rotation with respect to the nut or bolt head.

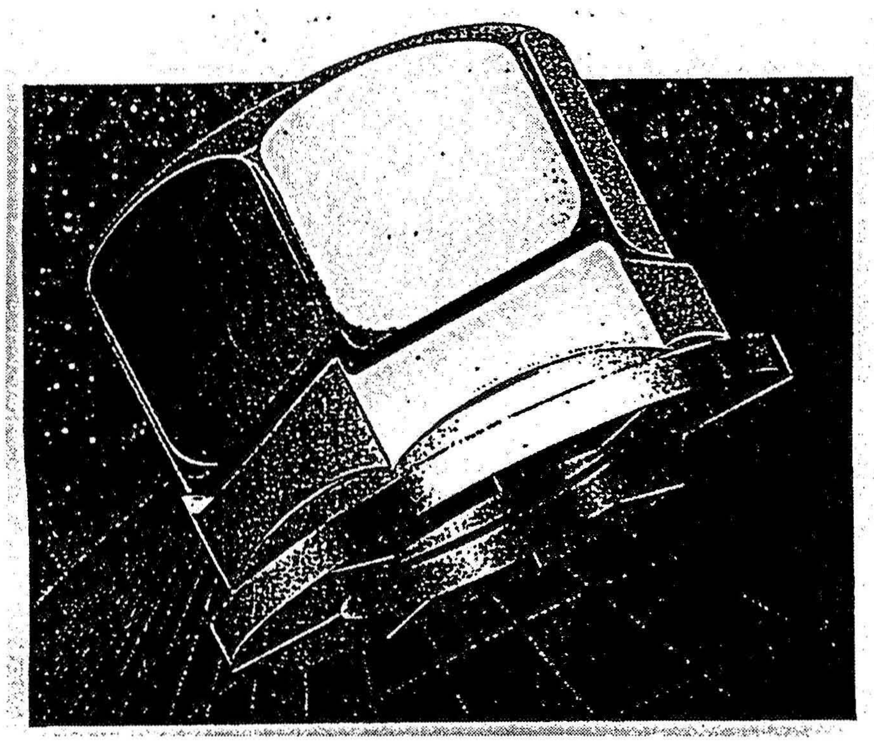
The ramps on the mating washers have a steeper angle than the thread helix angle of the fastener. When the nut or bolt tries to rotate, the steep ramp angle won't allow it to turn.

Damage tolerance of coated surfaces must be considered when using these washers.



LOCKING METHODS

- **Disk-lock KEP Nuts**



KEP Nut

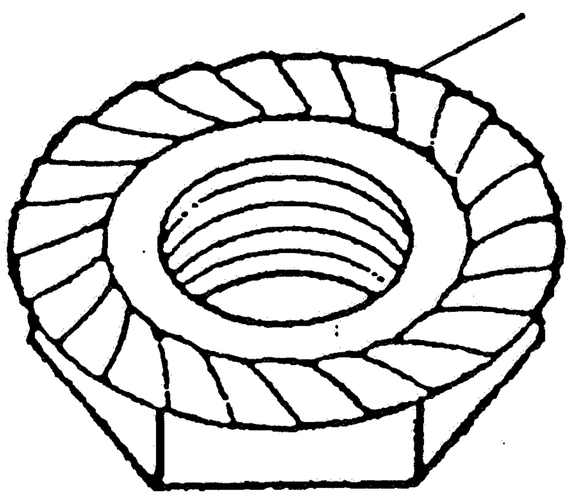
- The locking principle of this nut is the same as that of the Disc-lock washers. The difference here is that the bottom half of the nut is unthreaded with locking serrations on the bottom.

- The two nut halves are installed as a unit with a socket. The ramp angle prevents loosening of the threaded half with respect to the unthreaded half.

- These nuts are frequently used for heavy truck wheels.

LOCKING METHODS

- Serrated Nut or Bolt Head



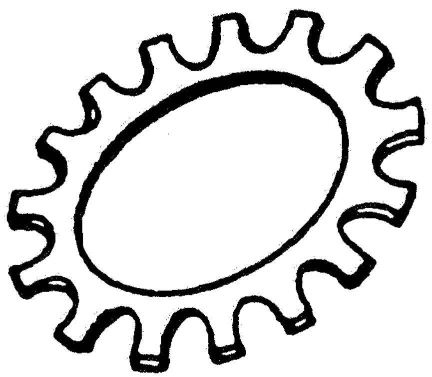
Durlock™ Nut

- This type of nut or bolt also depends on embedment of the serrations in the contact surfaces for locking.

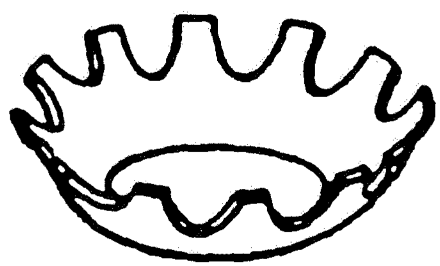
- Damage tolerance of coated surfaces is also a consideration for this nut/bolt configuration.

LOCKING METHODS

- Tooth Lockwashers



Flat

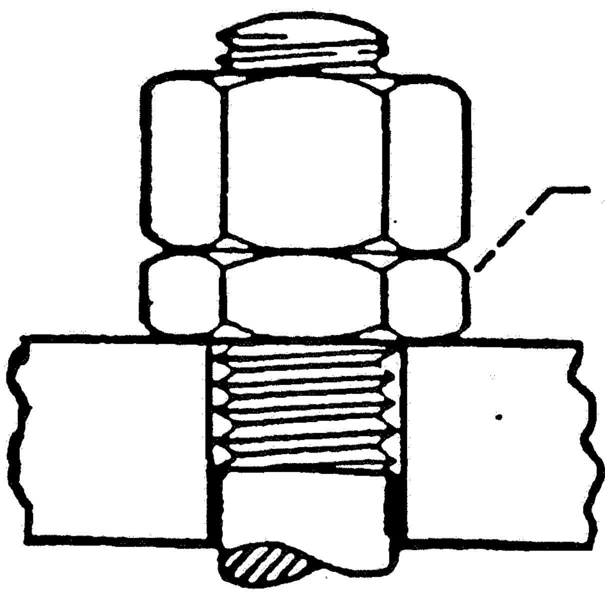


Countersunk

- Tooth lockwashers are used to lock heads or nuts to their attaching surfaces.
- The teeth are formed in a twisted configuration with sharp edges.
- One edge bites into the bolthead (or nut) while the other edge bites into the mating surface.
- The flat tooth washer is also available with internal teeth.
- Surface finish is damaged by the washer teeth.

LOCKING METHODS

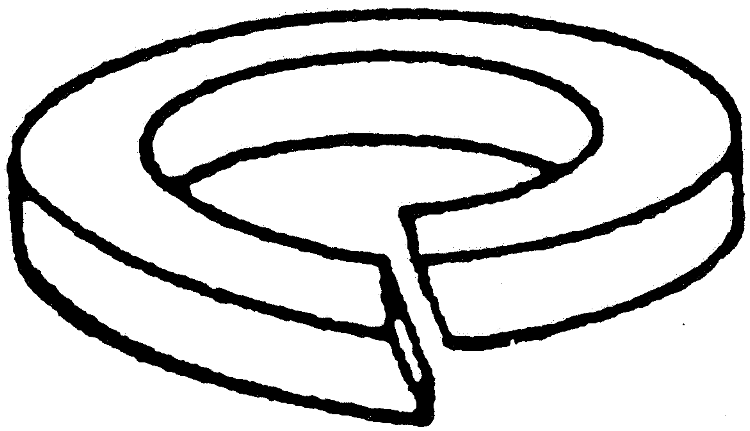
- Jam Nut



- These thin nuts are normally "jammed" into a regular nut.
- The "experts" can't agree on whether the jam nut is on the bottom (shown at left) or top.
- It is very difficult to load each nut so that they will work together to carry the total axial load. The best design rule would be to design the heavy nut to carry all of the design load.
- A jam nut should not be used for locking a critical fastener. However, it can be used for locking a turnbuckle, as the turnbuckle rod carries all of the axial load.

LOCKING METHODS

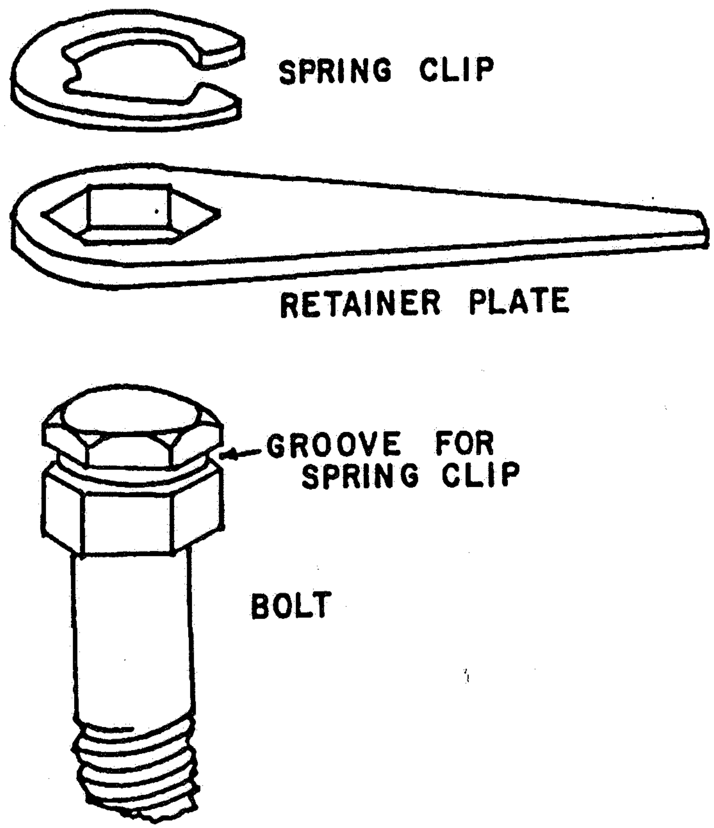
- Split (Helical) Lockwashers



- Split lockwashers provide a slight amount of locking capability, but they flatten under normal tensile load for the fastener.
- Vibration testing of split lockwasher assemblies indicate that they are about the same as a flat washer to loosen under vibration.
- Split lockwashers are not recommended as a locking device for any fastener locking application.

LOCKING METHODS

- Stage 8™ Fastening System



- This system consists of a special grooved bolt head, a spring clip, and a retainer plate (as shown at left).
- The retainer plate is fitted over the bottom of the bolt head after bolt installation. The plate is locked in place with the spring clip.
- The retainer plate must have a protruding surface or component to lock up against in order to prevent bolt rotation.
- One drawback to this type of system is that the retainer plate is usually thin sheet metal which will rust away in a corrosive environment.

WASHERS

- Introduction

  Most washers are flat and are used to provide a hardened smooth surface for contact of the fastener head or nut.

  They are used as spacers under nuts in shear joints to tighten the nut while maintaining smooth shank in the hole.

  Washers usually prevent embedment in the joint material by a bolthead or nut, thus making the bolt tension (from torque) more predictable.

  A washer can be hit laterally and broken loose in a rusted joint, whereas a bolt or nut is harder to pound loose.

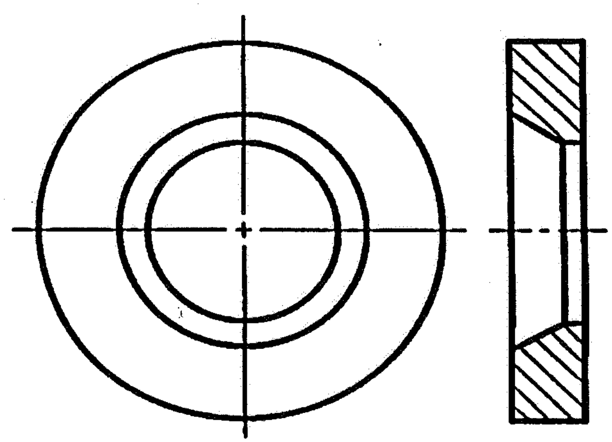
WASHERS

- **Plain Flat and Countersunk**

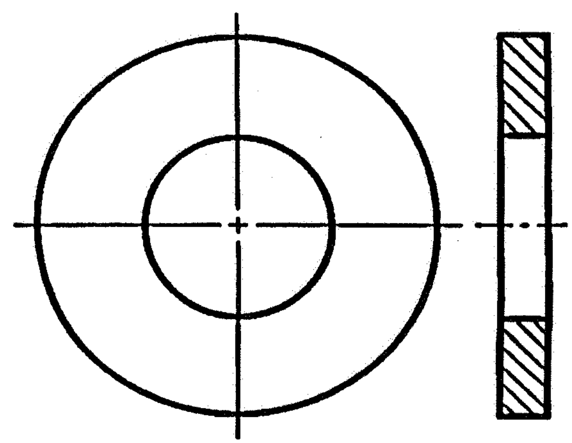
These are usually called out by a standard (MS, AN, ANSI, etc.) which defines the outside diameter, inside diameter and thickness for a given size fastener.

Countersunk washers are used to clear the head radius on a high strength bolt.

Custom washers with variable outside diameters can be special ordered.



COUNTERSUNK WASHER
(For use under bolt head)



PLAIN WASHER
(For use under nut)

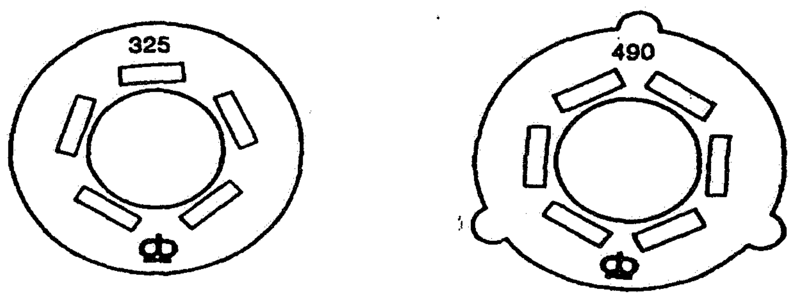
WASHERS

See Locking Methods Section for the washers listed below.

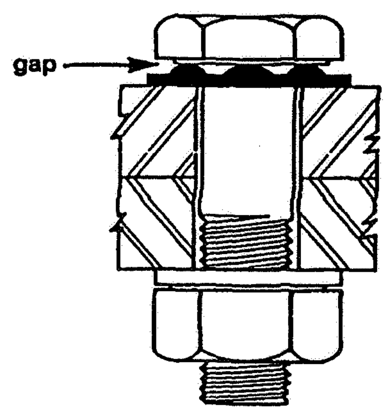
- Split helical lockwasher
- Toothed lockwasher
- Disc-lock washer

WASHERS

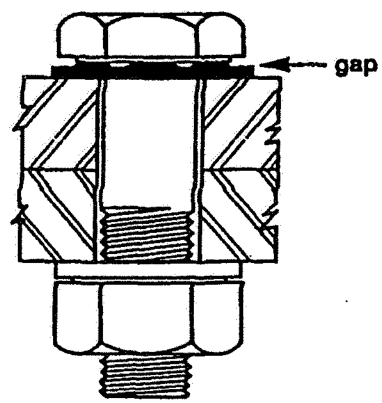
- **Direct Tension Indicating (DTI) to Measure Bolt Axial Load**
  - ➛ **Flat round washers that are stamped with 5 or 6 "bumps" on the top surface (as shown below).**
  - ➛ **A regular washer is placed between the bolt head or nut to allow for uniform compressing of the bumps.**
  - ➛ **The bolt tensile load vs. bump compression is determined by the washer manufacturer. The bolt is torqued until the proper gap is reached (inspected by feeler gage) for the desired load.**



DTI Washers



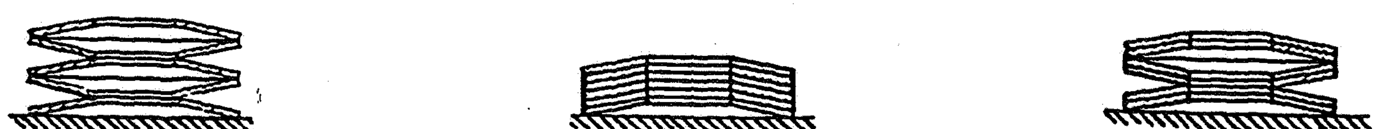
No load



Fully Torqued

WASHERS

- **Belleville Washer**
  - ➛ Patented in 1867 in Paris, France by Julien F. Belleville.
  - ➛ This washer is also known as a cone washer or spring washer.
  - ➛ It can be used as a load limiting device for fastener tensile load. Each washer has a given design load necessary to elastically compress it flat.
  - ➛ It is also used as a “spring” for absorbing differential thermal expansion between the fastener and joint materials.
  - ➛ Belleville washers may be used in stacks (parallel or in series) to establish a given spring constant for compression or to increase the working length of the fastener.



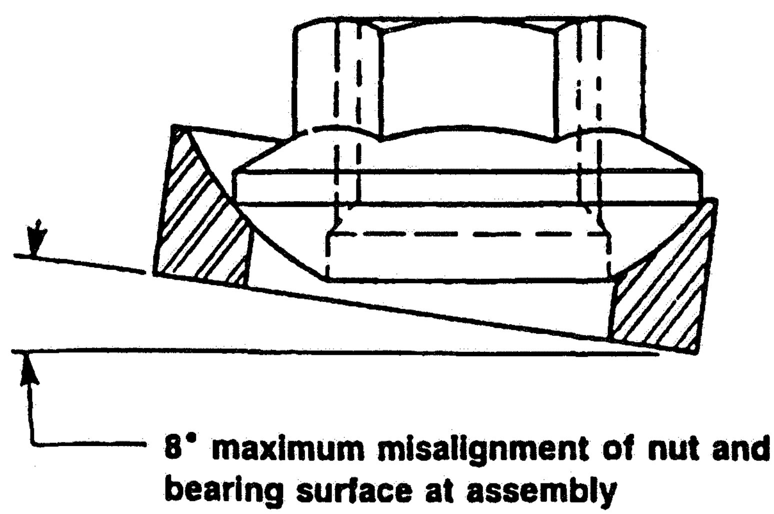
(a) In series. (b) In parallel. (c) In parallel series.

Combinations of Belleville washers.

WASHERS

- **Self-Aligning**

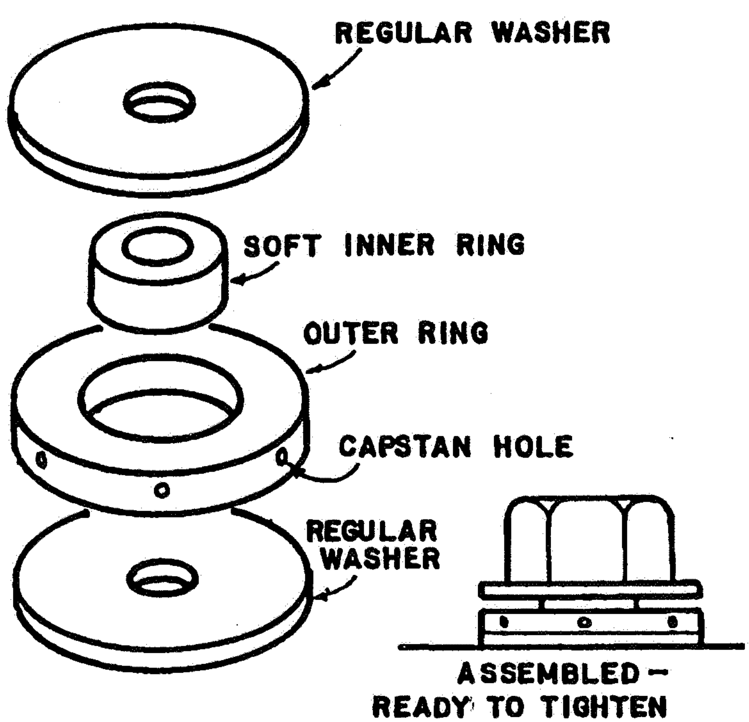
  - This washer and nut are machined as an assembly.
  - These washers are sometimes used on structural shape flanges.
  - The use of this assembly should be avoided, if possible, due to cost and weight considerations.



Self-Aligning Nut/Washer Assembly

WASHERS

- Preload Indicating (by SPS)



- This washer is a four-part assembly (as shown at left). There are two outer plain washers, with a soft inner ring and an outer ring with capstan holes.

- The inner ring compresses under load until it expands to fill the internal diameter of the outer ring and reaches the same height as the outer ring. (At this point the outer ring should not turn.)

- The inner rings are color coded to indicate their load ratings.

INSERTS

- Introduction

An insert is a special bushing that is threaded on its inside diameter and locked with threads or protrusions on its outside diameter in a drilled, molded, or tapped hole. It is used to provide a strong, wear-resistant tapped hole in a soft material such as plastic and nonferrous materials, as well as to repair stripped threads in a tapped hole. They usually are coated to resist corrosion and seizing.

In general, there are two types of inserts: those that are threaded externally, and those that are locked by some method other than threads (knurls, serrations, grooves, or interference fit). Within the threaded inserts there are three types; the wire thread, the self-tapping, and the solid bushing.

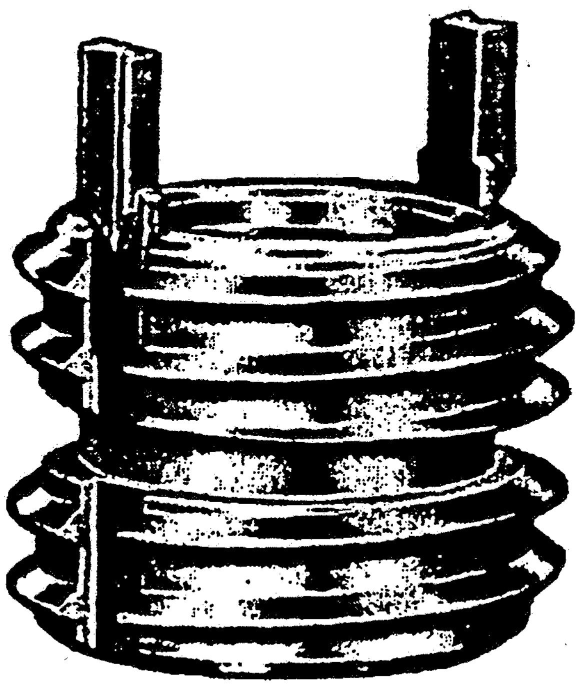
INSERTS

- **Introduction (Cont'd)**

  The aerospace industry uses inserts in tapped holes in soft materials in order to utilize small high-strength fasteners to save weight. The bigger external thread of the insert (nominally 1/8 in. bigger in diameter than the internal thread) gives, for example, a 10-32 bolt in an equivalent 5/16-18 tapped hole.

INSERTS

- Solid Bushing (Keensert™)

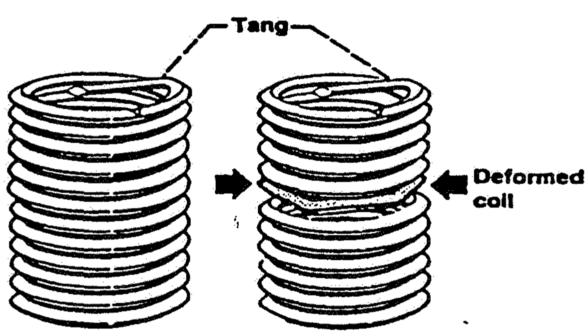


Locking Keensert

- Keensert is the generic name for a solid externally threaded insert.
- This insert will usually have external locking tangs (two or four, depending on the size of the insert). They are also available with internal deformed thread locking.
- Keenserts are available in different materials, coatings, and service ratings (light, normal, and heavy duty).
- Installation of Keenserts is labor intensive, so they should not be used unless required.

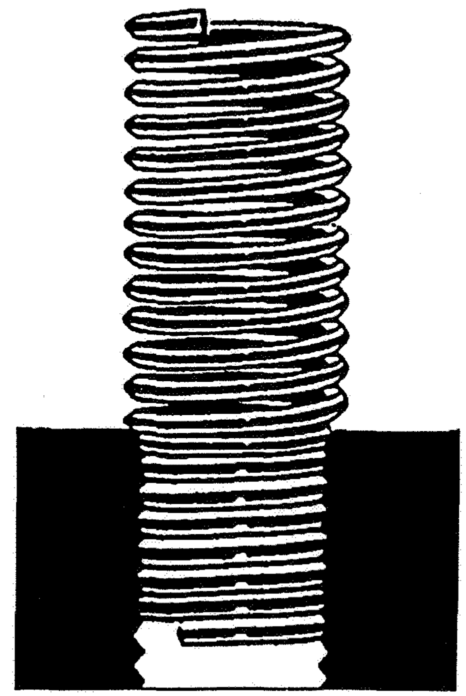
INSERTS

- **Wire Thread (Helicoil™)**
  - ➣ Helicoil is the generic name for a wire thread insert.
  - ➣ The helicoil wire is usually made of CRES and has a diamond-shaped cross section that forms both internal and external threads when installed (as shown below).
  - ➣ Helicoils are usually coated to deter corrosion and seizing
  - ➣ Available as locking and non-locking
  - ➣ Tang is broken off after installation.
  - ➣ Available in lengths of 1D to 3D.
  - ➣ Widely used for repair of stripped threads



(a) Free running (b) Locking

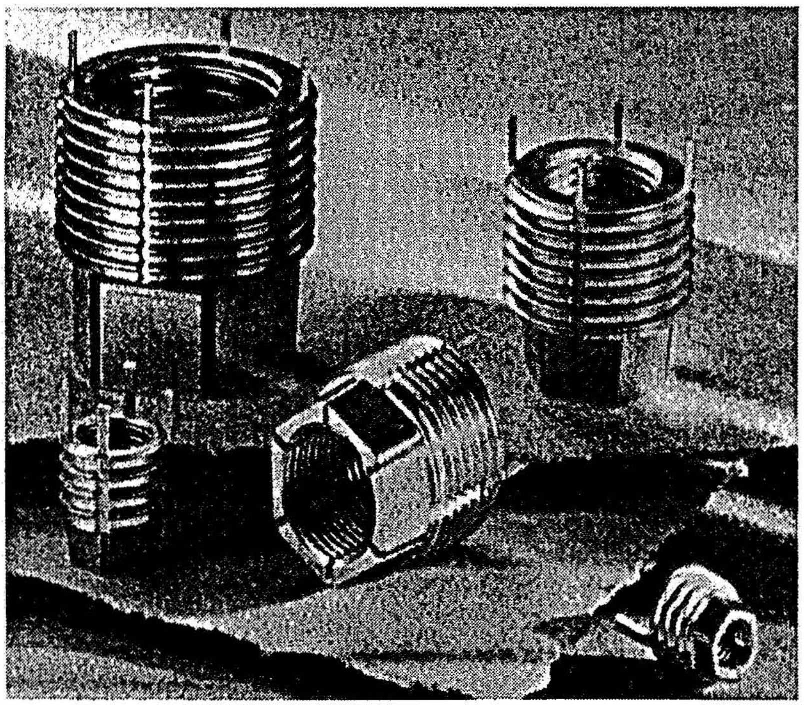
Wire thread insert types



Wire thread insert installation

INSERTS

- Split-beam Locking (FlatBeam™)

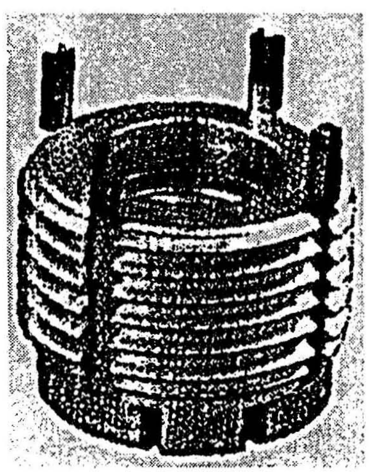


- This is a regular Keensert with a split-beam locknut machined on the bottom (as shown at left).
- These inserts are made of A-286 or Inconel 718 but the external thread length is slightly less than comparable size Keenserts. Tension loads in the soft material would have to be checked to avoid pull-out problems.
- Adding the split-beam feature to the insert increases its overall length, so this insert requires about 15% more depth of drilled hole for a given size.

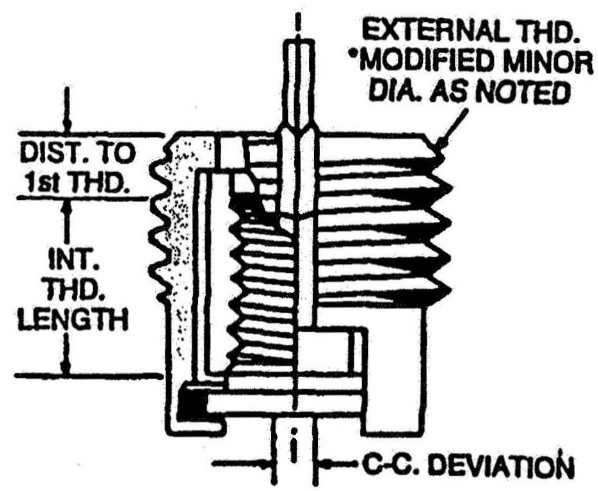
INSERTS

- ## Floating

  - The floating insert is manufactured in two pieces with a "floating" internal threaded insert which can move with respect to the externally threaded part to take up misalignment of the mating fasteners.
  - This insert is recommended when countersunk fasteners are installed to avoid head bending.
  - For a given size internal thread, the external thread must be larger than for a one-piece insert.



Floating Insert



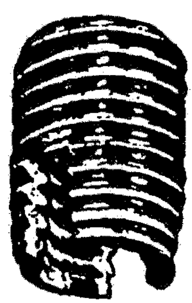
INSERTS

- Self-Tapping

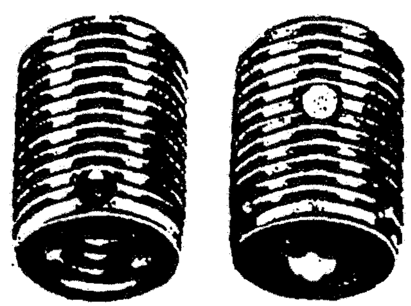
  - ➛ This is usually a solid bushing with tapered external threads like a self-tapping screw.

  - ➛ This insert is usually non-locking or locked with a Nylok™ pellet.

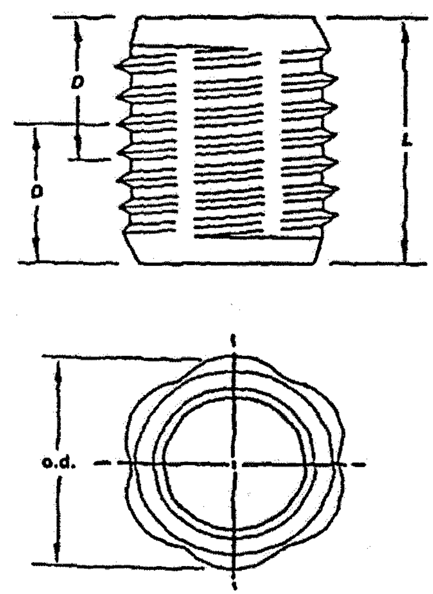
  - ➛ A Speedsert™ insert has a lobed thread which deforms the (soft) material without cutting chips.



Slotted



Nylok



Speedsert

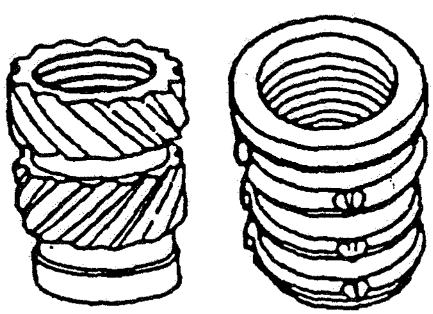
INSERTS

- ## Unthreaded External Diameter

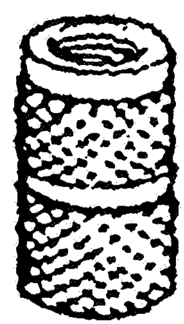
  ➛ Ultrasonically installed - This insert is installed in plastic material by applying force with ultra-sonic vibration to melt the hole area. The insert is then molded in place as the plastic cools.

  ➛ Molded-in-place - This insert is mounted in the mold before the plastic is cast. The insert becomes an integral part of the plastic component.

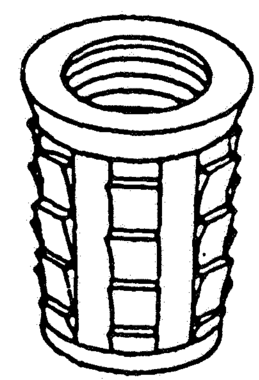
  ➛ Plastic expandable insert - This is the ordinary household type insert for drywall mounting of shelves, etc.



Ultrasonic Inserts



Molded-in-place Insert



Plastic Expandable Insert

INSERTS

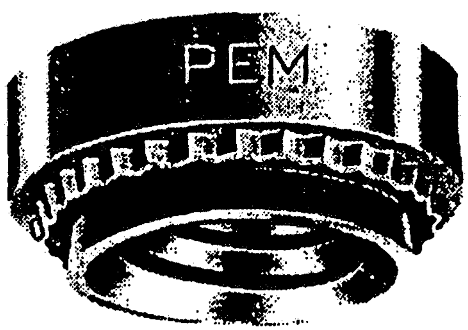
- **PEM™**

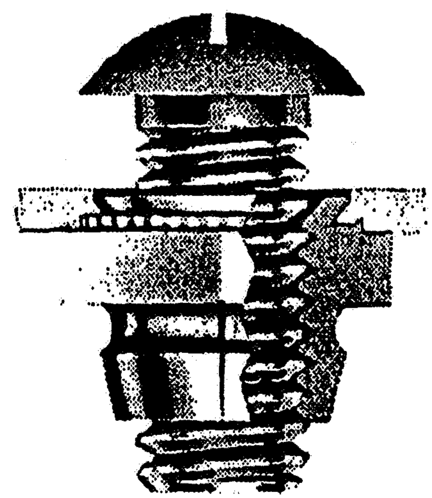
  ➢ This fastening device is a cross between an insert and a nut. It locks in place on the back side of the sheet with serrations that bite into the edges of the hole.

  ➢ PEM nuts are usually used for thin sheet installations where there isn't enough depth for an insert.

  ➢ PEM nuts are not recommended for structural applications, due to their unpredictable clinching capabilities and damage to their mounting surfaces.

  ➢ Two examples of PEM nuts are shown below.





NUTPLATES

- Introduction

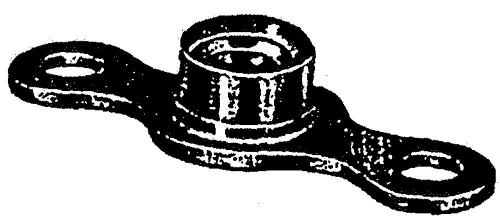
A nutplate (anchor nut) is normally used as a blind nut. It can be fixed or floating. In addition, it can have most of the locking and sealing features of a regular nut. Note that a floating nutplate *DOES NOT* provide for angular misalignment of fasteners.

Nutplates are normally used on a sheet that is too thin to tap. They are used primarily by the aerospace companies, since three drilled holes and two rivets are required for installation.

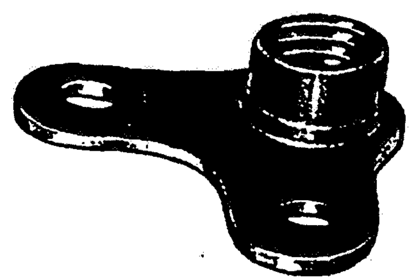
NUTPLATES

- Common Nutplates (courtesy of Kaynar)

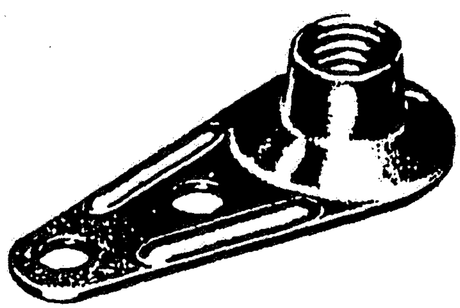
  ➔ Fixed
  - Two-lug
  - Mickey Mouse (corner)
  - One-lug
  - Sealed



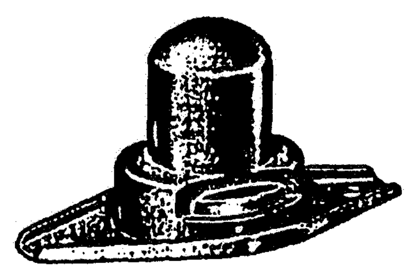
Two-lug



Mickey Mouse (corner)



One-lug

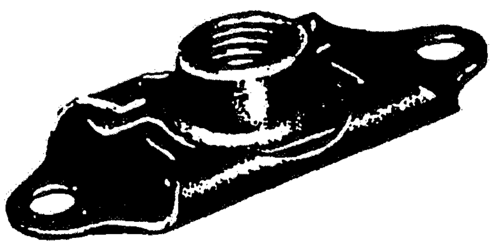


Sealed

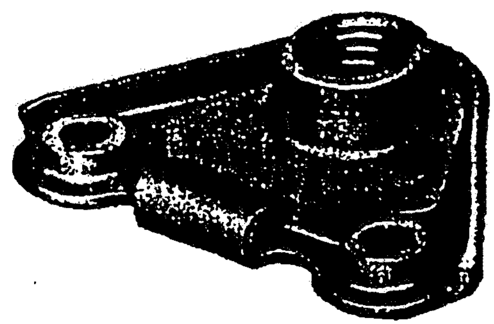
NUTPLATES

- **Common Nutplates** (Cont'd)

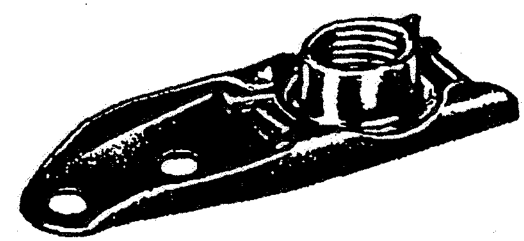
  ➛ Floating
  - Two-lug
  - Mickey Mouse (corner)
  - One-lug
  - Sealed
  - Replaceable Nut



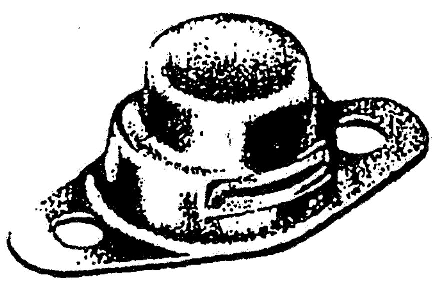
Two-lug



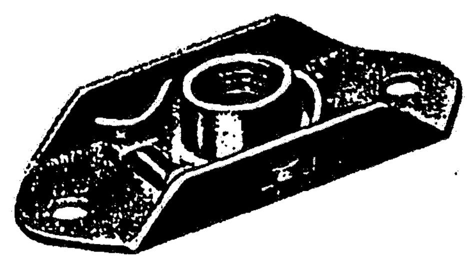
Mickey Mouse (corner)



One-lug



Sealed



Replaceable Nut

THREADS

- ## Introduction

Geometric information on most threads can be found in American National Standards Institute (ANSI) and National Institute of Standards and Technology (NIST) Federal Standard H-28, so no thread tables will be given here. My goal is to explain the different types of threads and taps and why the different types are used.

The most common (inch) threads are unified national coarse (UNC) and unified national fine (UNF). Other threads used are unified national extra fine (UNEF), UNJC, UNJF, UNR, UNK, and constant pitch threads. Cut or ground threads in many of the above are also common.

THREADS

- **Introduction** (Cont'd)

  Metric fasteners are also available in coarse, fine, and J-threads (M and MJ). Although metric fasteners are also covered by ANSI and H-28 specifications, the aerospace metric fasteners are covered by NAXXXX and MAXXXX specifications, which are issued by the Aerospace Industries Association (AIA), a branch of the Society of Automotive Engineers (SAE).

THREADS

- Definitions (see Figure 5(a) for illustrations of definitions)
  - *"Pitch"* is the center to center distance between adjacent threads.
  - *"Major diameter"* is the largest diameter of the threads (marked "P").
  - *"Minor diameter"* (also root diameter) is the minimum diameter from root radius to root radius.
  - *"Vanish cone"* is a thread run-out defined by an angle (12° to 35°).

THREADS

- **Definitions** (Cont'd)

  - ➢ *"Crest"* is the top of the thread (at major diameter).

  - ➢ *"Flank"* is the flat portion of the thread between the root radius and the crest radius.

  - ➢ *"Pitch diameter"* is the theoretical diameter where the shear thickness of mating threads is equal. (It is usually very close to the average of the major and minor diameters.)

  - ➢ Further definitions are found in ANSI B1.1 and Fed Std. H-28.

THREADS

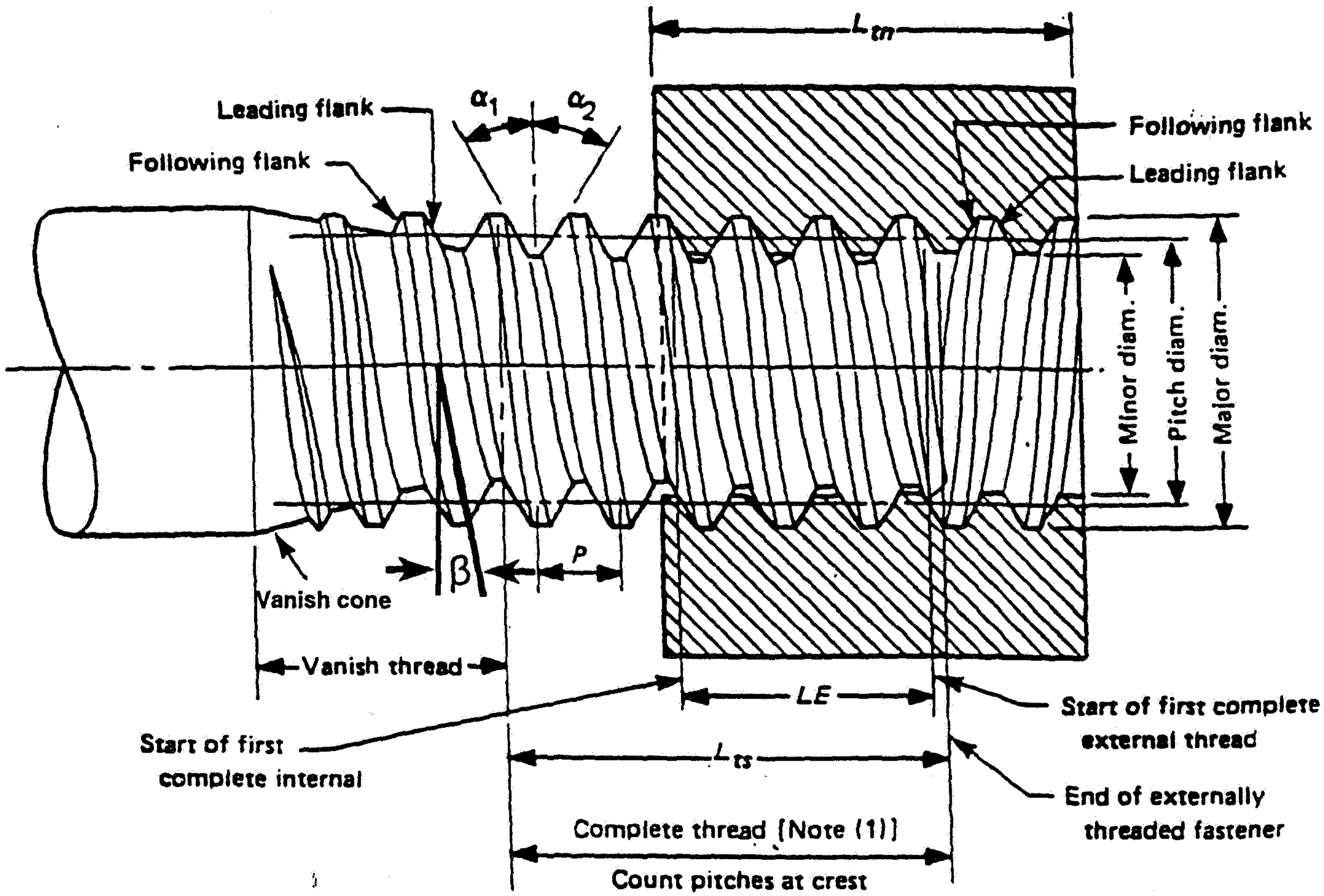


Figure 5(a) - Thread Terminology

THREADS

- **Thread Pitch and Lead**

As mentioned earlier, the thread pitch is the distance between adjacent threads. In the inch system this distance is given as threads per inch of fastener length such as a 1/4-20 (1/4 inch diameter with 20 threads per inch).

In the metric system, the pitch is given directly.

The lead of a screw thread is the distance the nut will move forward on the screw per one 360° nut rotation. In a single lead thread the pitch and lead are the same. For a double lead the nut would move forward two threads per 360° rotation. Structural fasteners are single lead; double (or triple) lead thread might be found on a bleach bottle.

THREADS

- ## Rolled External Threads

The "R" in UNRC or UNRF indicates rolled threads. Rolling is the method used by most external thread producers. The bolt or screw threads are cold-rolled in automatic machines up to about .75 inch diameter bolts. Larger diameter threads require some pre-heating to allow proper thread rolling. The bolt stock (wire coil) is annealed prior to making the bolts to ensure that it is as soft as possible before forming begins.

Heat treating of commercial fasteners is usually done *AFTER* all forming operations are completed.

High strength (above 180 ksi) fasteners are usually heat treated *BEFORE* threading to make the threads more fatigue resistant and stronger. Cold-rolling the threads puts compressive residual stresses in the thread surfaces and raises the overall strength of the material through cold work.

THREADS

- ## UNC Threads

UNC threads are the most common ones for general-purpose fasteners. Coarse threads are deeper than fine threads and are easier to assemble without cross threading. Nearly all fasteners up through 0.164 in. dia. are UNC. UNC threads are easier to remove when corroded, due to their (usually) sloppy fit. However, UNRC or UNC fasteners can be procured to tighter (class 3) fits if needed. (Classes of fit will be covered later.)

- ## UNF Threads

UNF threads are the standard for the aerospace industry. Almost all fasteners with strengths above 150 ksi are UNRF or UNJF. UNF threads have a slightly larger minor diameter, giving a larger net cross-sectional area than UNC fasteners with the same external diameters. The smaller lead angle of the threads allows better torque adjustment.

THREADS

- ## UNEF Threads

  UNEF Threads are used for tapped holes in thin materials or hard materials, as well as for holes with short edge distance. They are also used on adjusting screws for mechanisms.

- ## UNR and UNK Thread Comparison

  - UNR threads are the same as rolled UN threads, except that the root radius must be rounded. However, the root radius *IS NOT* measured or toleranced. Most UNF and UNC external threads are rolled in this manner in automatic machines. There is no internal UNR thread.

  - UNK threads are the same as UNR threads, except that the root radius *IS* measured and toleranced. UNK external threads are not common. There is no UNK internal thread.

THREADS

- ## Constant Pitch Threads

Constant Pitch threads are only used to tailor a design to a particular need for diameters up to approximately one inch. For diameters above one inch, regular threads become very difficult to form. Going to constant pitches of 8, 12, or 16 threads per inch allows the threads to be rolled (at elevated temperatures). The torque-tension adjustments are also easier to apply with the finer thread.

- ## Left Hand Threads

Left hand threads do exist (e.g. gas grill tanks and turnbuckles). The heads of the bolts are sometimes marked and LH nuts may have external notches to identify them.

THREADS

- UNJF Threads

UNJF threads are made in both external and internal forms, but the external form is the most common. “J” threads have a much larger root radius than UNF or UNC threads, and inspection of this radius is mandatory. Since the root radius is larger, the minor diameter is larger, giving a higher net cross-sectional area. This larger root radius also reduces the stress concentration factor in the threaded area. Therefore, high strength bolts (above 180 ksi) usually have J-threads.

Although some nuts have J-threads, there are no J-taps for internal holes.

THREADS

- ## UNJC Threads

UNJC threads are coarse threads with the same characteristics listed above for UNJF threads. However, UNJC threads are not common, except in small sizes up through a No. 8 (.164 dia.).

Note that a “J” bolt requires a “J” nut. However, a “J” nut will fit a regular bolt of the same size thread. Also note that a “J” nut has rolled threads, but a “J” tapped hole is a misnomer. (Ref. ANSI B1.1, Appendix A). A normal tapped hole has *CUT* threads, regardless of the type of thread form.

THREADS

- Internal Threads
  - ➢ Internal threads are usually cut with a tap, although some nut threads are rolled.
  - ➢ Tapped holes are more difficult to inspect than external threads, so their design margins should be higher than those for external threads.
  - ➢ There is no J-thread form for tapped holes.

THREADS

- ## Thread Classes (inch)

Thread classes are distinguished from each other by their dimensional tolerance ranges for a given size. The class designations run from 1A to 3A and 1B to 3B for external and internal threads, respectively. A Class 1 is a loose fitting general (hardware store variety) thread. A Class 2 is an industrial quality fit and a Class 3 is a close fit. The aerospace industry uses Class 3 threads.

A complete tabulation of thread sizes and classes is given in Federal Standard H-28, as well as the SAE Handbook.

THREADS

- Cut or Ground Threads

Cut or ground external threads are normally made on "one-of-a-kind" fasteners for a specific application. However, tapped holes are cut, unless the tapped material is soft enough that the tap can cold-form the threads.

From a structural perspective, cut threads are weaker than rolled threads, since thread cutting destroys the grain flow lines of the formed rod.

THREADS

- **Metric**

Metric threads are described as M(dia.) x pitch (e.g. M8 x 1.0 for a 8 mm diameter with 1.0 mm thread pitch). The pitch value tells whether the thread is fine or coarse for a given diameter. It won't be designated as a MC or MF in the callout. The only further identifier is MJ for metric J-threads. This means that the user has to look up the size and pitch combinations in a table to determine if the thread is coarse or fine. Also remember that the material property class of the fastener must be specified to tie down the strength requirements.

THREADS

• Metric Thread Classes

Metric threads have two diameter tolerances. The first is for the pitch diameter and the second is for the crest diameter. (Crest diameter is the major diameter for external threads and the minor diameter for external threads.)

If only one value is given, the tolerance for both diameters is the same. Metric nomenclature is to use "g" for external pitch tolerance, and "H" for internal pitch tolerance. The numbers to indicate the tightness of tolerances can vary from 3 (tightest) to 9 (sloppiest), with 4 to 6 being the normal tolerance ranges. Two number/letter values are given when the pitch and crest diameters have different tolerances.

THREADS

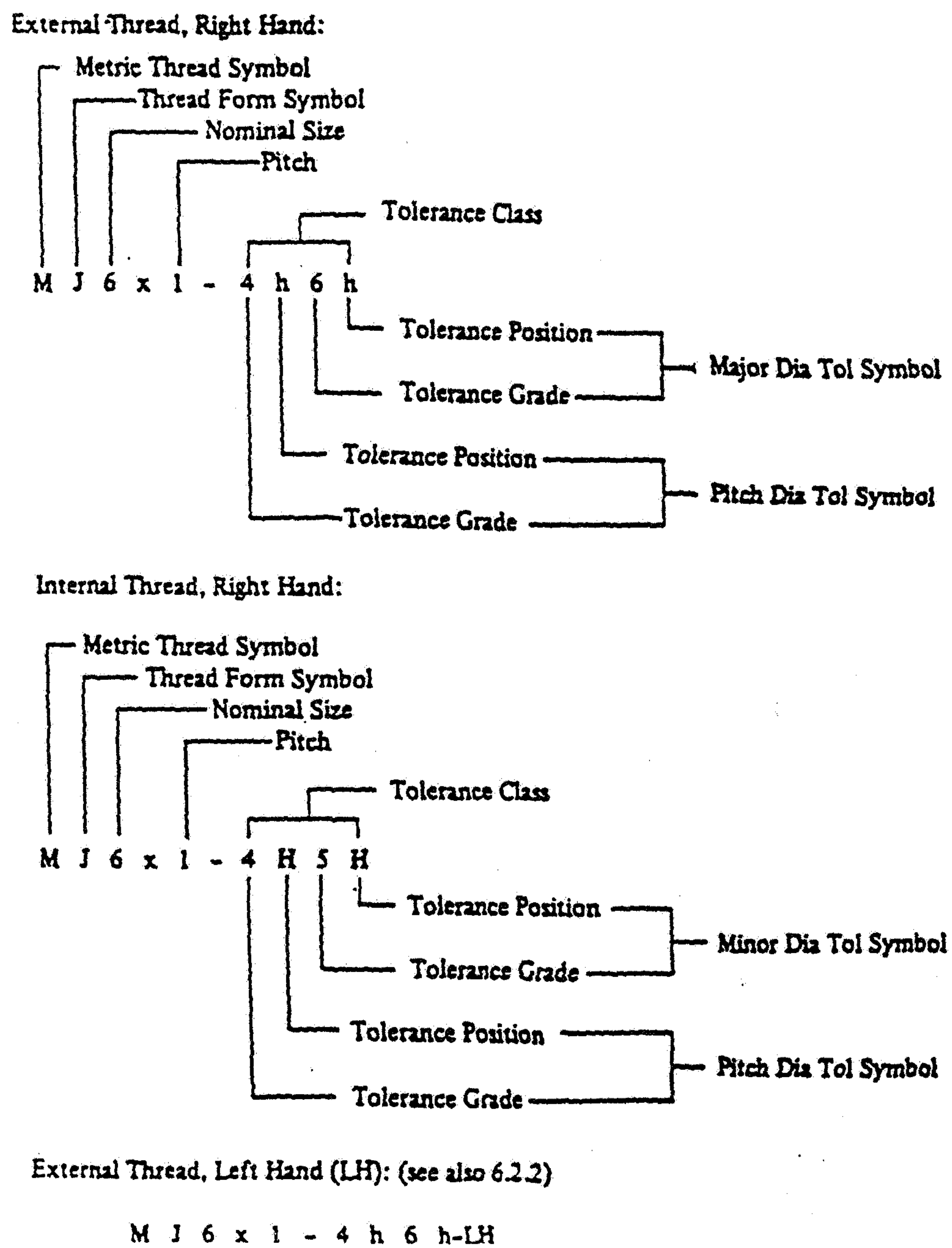
- **Metric Thread Classes** (Cont'd)

Sample callouts of metric MJ threads (from ANSI Y14.6) are given in Figure 6. Sample callouts of Metric M threads (from ANSI B1.3M) are given in Figure 7. Note that MJ uses "h" and M uses "g" for external thread tolerances.

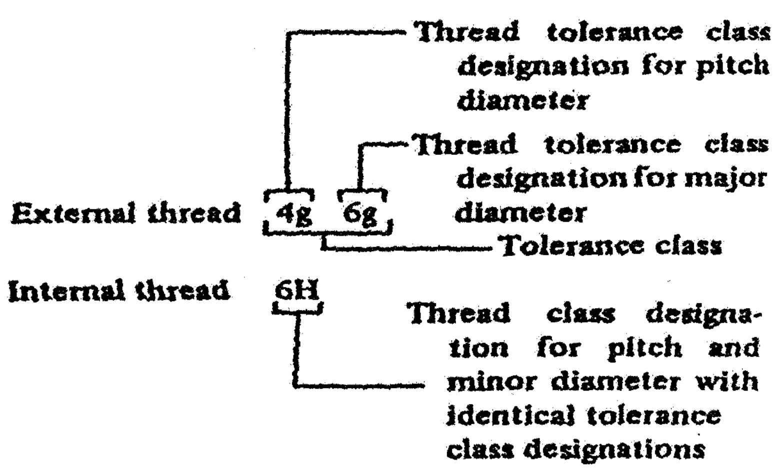
A comparison of inch vs. Metric thread classes is given in Table 9.

THREADS

Figure 6 - MJ Metric Thread Tolerances



THREADS



Note: *Unless otherwise specified in the designation, the screw thread helix is right hand.*

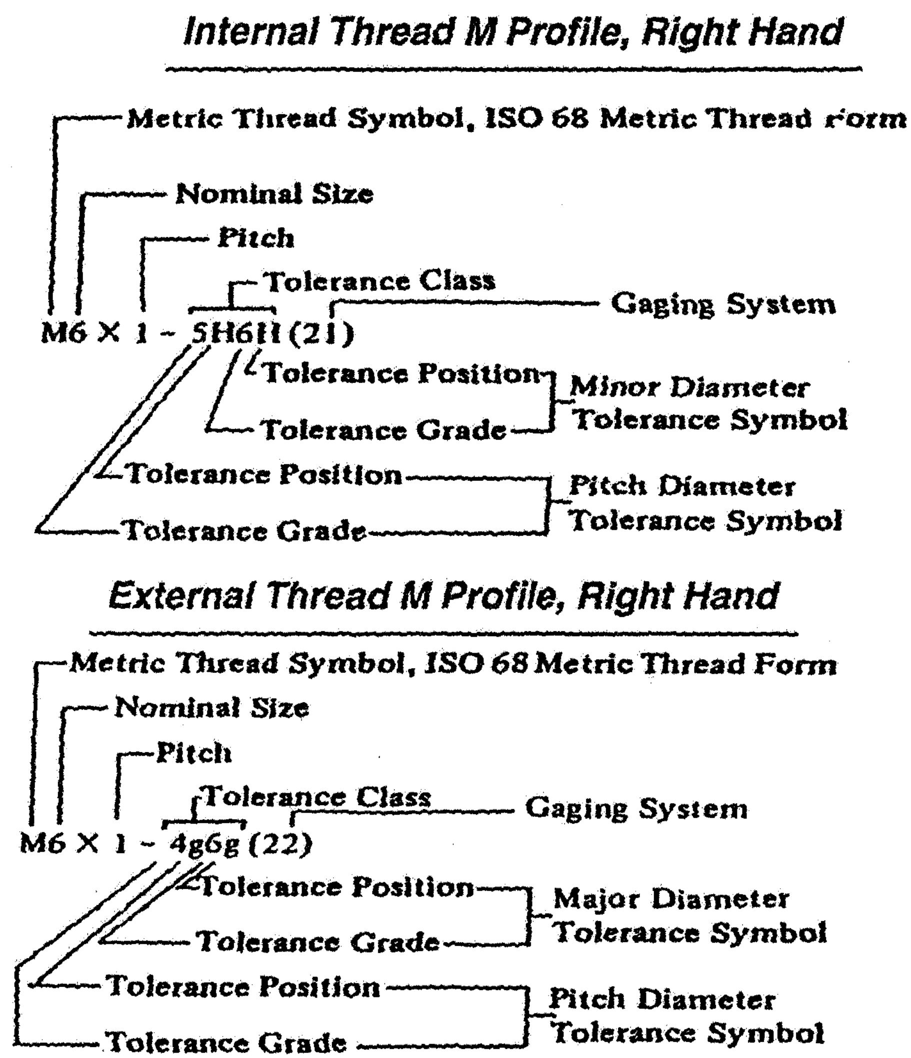


Figure 7 - M Metric Thread Tolerances

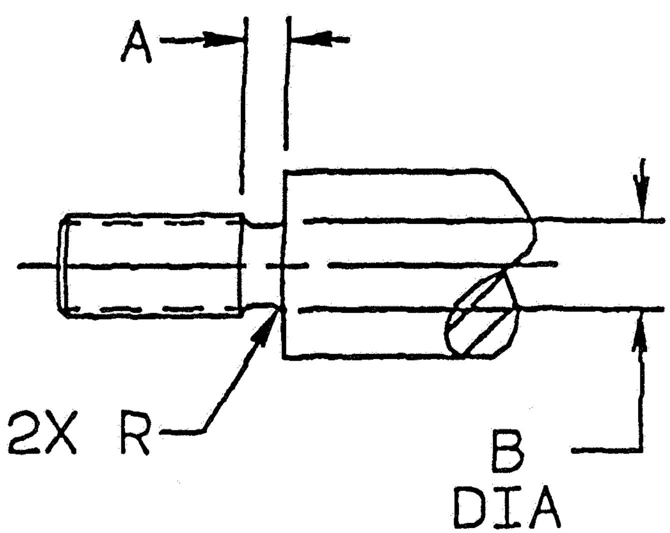
THREADS

Table 9 - Comparison of Inch to Metric Thread Tolerance Classes

Inch	Metric
2A	6g
2B	6H
3A	M: 4g6g MJ: 4h6h
3B	M: 4H5H MJ: 4H6H (≤ 5 mm) 4H5H (> 5 mm)

THREADS

- **Thread Relief**



External Thread Relief

Where a fastener is necked down (such as a shoulder bolt) some thread relief must be used (as shown at left). A complete table of values for the dimensions shown are given in ANSI B18.3 (for shoulder bolts).

Note that the "B" diameter should never exceed the minor diameter of the threads.

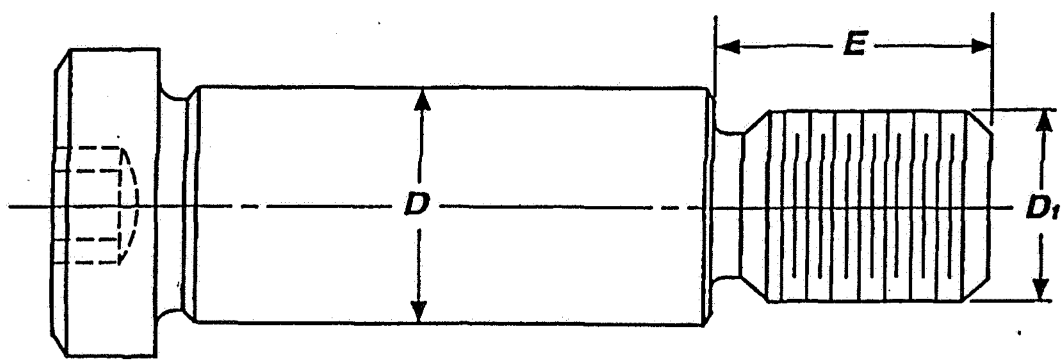
Internal thread relief is sometimes used where a through hole is partially tapped.

THREADS

- ## Shoulder Bolt

Since the previous page showed the thread relief for a shoulder bolt, it can be covered here.

Shoulder bolts are usually used as an actuator pin that is attached to a tapped hole in a mounting plate. The weakness of the shoulder bolt is that it will fail in bending in the thread relief area if it is loaded in single shear. The shank (D) must be supported to enable it to carry side (shear) loads.



Shoulder Bolt

THREADS

- ## Tapped Threads

Tapping holes must be done carefully to achieve the proper thread with the correct alignment. The tap drill sizes are given in several specifications, such as Machinery's Handbook, for common tap sizes in both inch and metric.

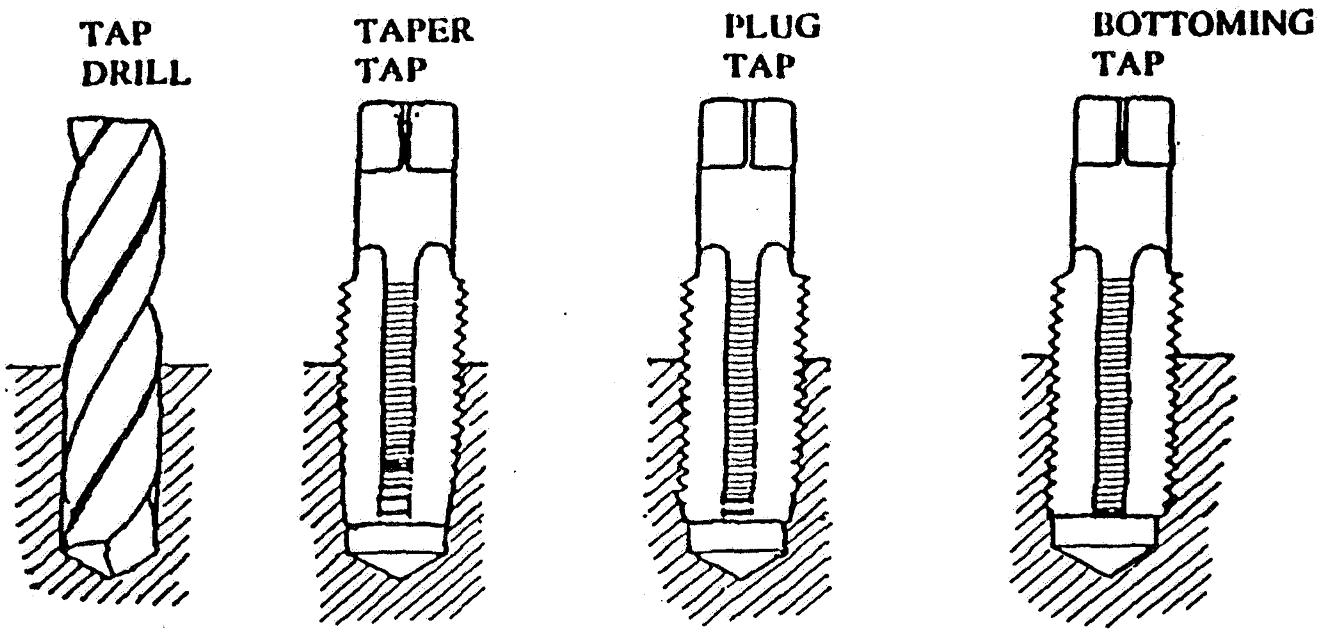
For larger diameter threads and fine pitch threads it is sometimes necessary to drill a slightly undersize hole and ream it to a more precise diameter prior to tapping.

The required length of tapped hole is dependent on the hardness of the material and the tensile load developed by the installed fastener. This length should also be based on complete threads, so final threading may require finishing the hole with a bottoming tap if total hole depth is limited.

An illustration of the different taps is shown in Figure 8.

THREADS

Figure 8 - Tapped Threads



Taper Tap - 7 to 10 Chamfered Threads
Plug Tap - 5 Chamfered Threads
Bottoming Tap - First Thread Chamfered Only

THREADS

- **Tapped Threads** (Cont'd)

Tapping may be accomplished by hand or by machine.

- The bulk of the metal in a threaded hole is removed by a tap drill which has a diameter equal to or slightly greater than the root diameter of the thread.
- The taper tap has from seven to ten chamfered threads along its axis. The long chamfer serves as a guide in aligning the axis of the tap with the hole.
- The plug tap has from three to five chamfered threads and is the most frequently used tap, particularly in machine tapping. In hand tapping it is not as easy to start as a taper tap.
- The bottom tap has a chamfer on the first thread only and should never be used as a starting tap. It is only used for finishing blind holes which require threading as close to the bottom as possible.

THREADS

- **ACME**

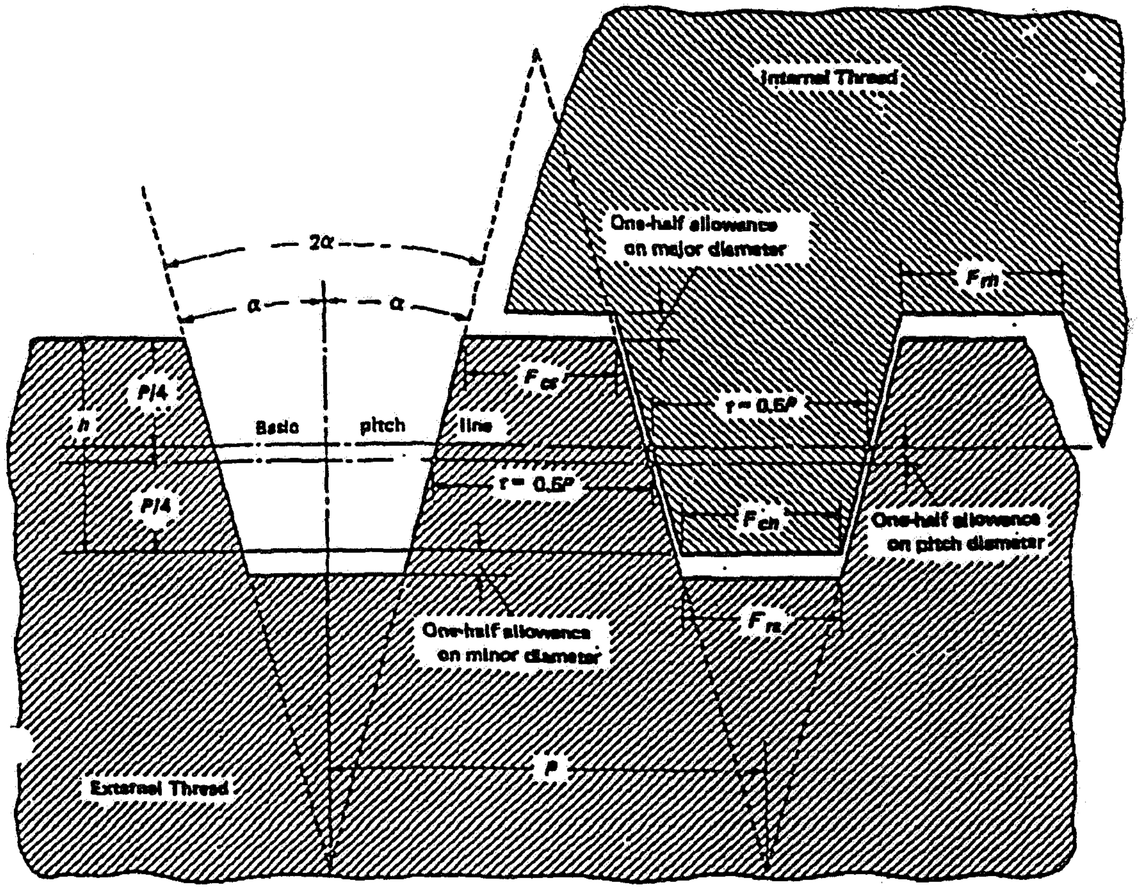
Acme threads have existed since the 1800's. They are used for both transmitting power (jacks) and traversing motions on machinery. The standard Acme thread is shown in Figure 9 (per ANSI B1.5).

Acme thread has general purpose fit classes of 2G, 3G, and 4G, with 2G being the sloppiest and 4G being the tightest.

Acme also has a preferred series of diameters and threads (centralizing) which are to be used whenever possible. These threads have similar tolerances, except that the three fit classes are 2C, 3C, and 4C, where "C" identifies "centralizing" (for better fit). Class 2C is the sloppiest and 4C is the tightest tolerance. The same tolerance designations are used for both internal and external threads.

THREADS

Figure 9 - Design Profile for External and Internal General Purpose ACME Thread



F_{cn} = basic width of flat of crest of internal thread
= $0.3707P$
F_{cs} = width of flat of crest of external thread
= $0.3707P - 0.259 \times$ pitch diameter allowance on external thread
F_{rn} = $0.3707P - 0.259 \times$ (major diameter allowance on internal thread)
F_{rs} = $0.3707P - 0.259 \times$ (minor diameter allowance on external thread — pitch diameter allowance on external thread)
P = pitch
h = basic height of thread
= $P/2$
n = number of threads/in.
t = thickness of thread
= $P/2$
α = 14 deg. 30 min.
2α = 29 deg.

THREADS

- ## STUB ACME

Stub Acme threads have existed since the early 1900's. The only difference in geometry between regular Acme and stub Acme threads is the height of thread. Regular Acme threads have a height of 0.5 pitch while stub Acme threads have a height of 0.3 pitch. The standard stub Acme thread is shown in Figure 10 (per ANSI B1.8). Stub Acme threads are normally used where heavy threads are required but total diameter is limited.

Stub Acme threads are usually available only with class 2G tolerances of ANSI B1.5, with classes 3G and 4G available only on special order.

THREADS

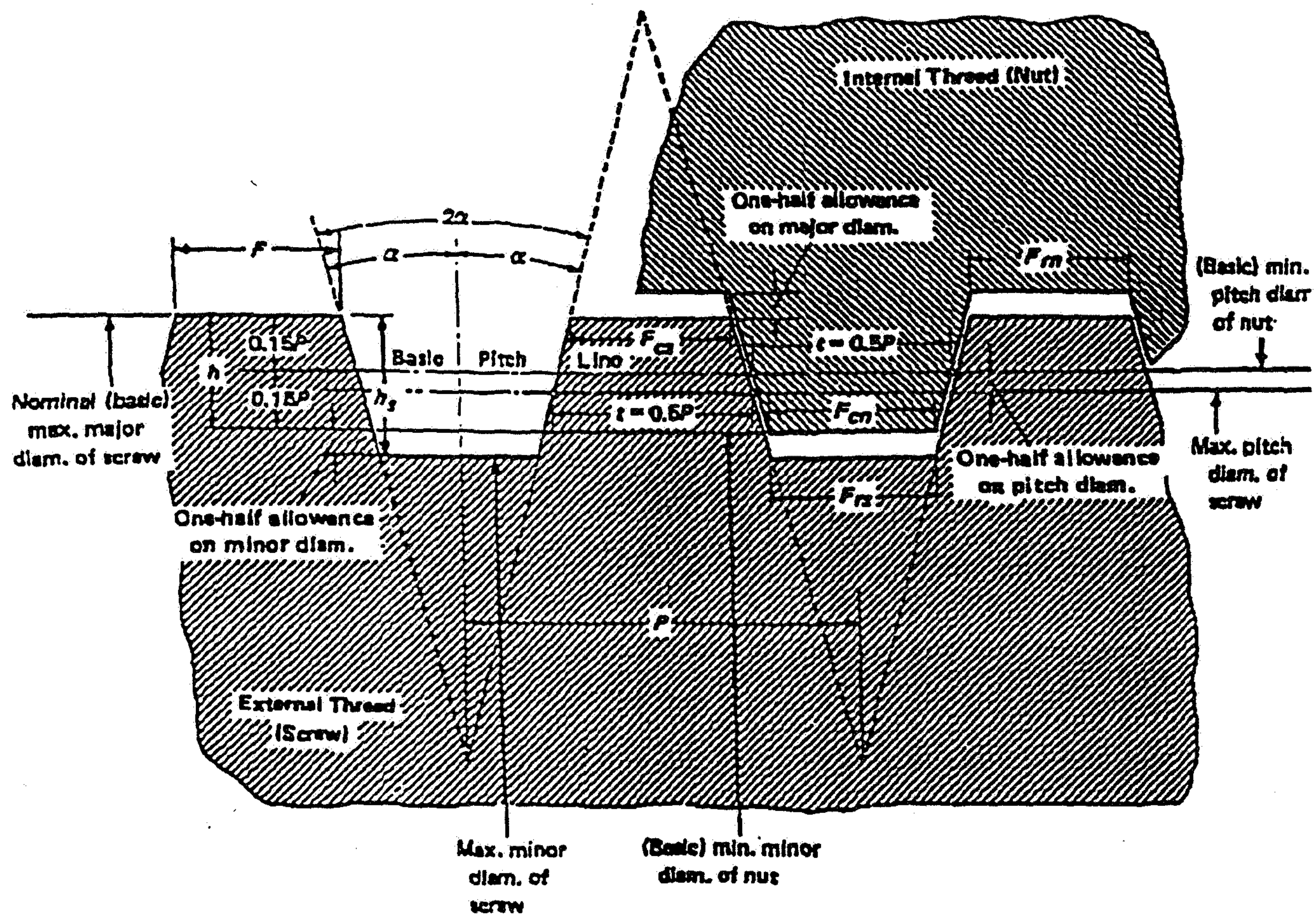


Figure 10 - Stub ACME Form of Thread

2α = 29 deg.
α = 14 deg. 30 min
P = pitch
n = number of threads/in.
N = number of turns/in.
h = 0.3P, basic thread height
F_{cn} = 0.4224P, basic width of flat of crest of internal thread
F_{cs} = 0.4224P − 0.259 × (pitch diameter allowance on external thread)
F_m = 0.4224P − 0.259 × (major diameter allowance of internal thread)
F_{rs} = 0.4224P − 0.259 × (minor diameter allowance on external thread − pitch diameter allowance on external thread)

THREADS

• Buttress

Buttress threads have been around since 1888, but they are special threads. They are used where loading is in one direction. Typical examples are airplane propeller hubs, columns for hydraulic presses, and breech assemblies of large guns.

This thread has a flat (7°) angle on the pressure flank and a 45° angle on the clearance flank. The height of threads (h) varies from 0.4 pitch (P) to 0.6 pitch with 0.6P being the preferred U.S. standard and 0.4P the preferred British standard.

The buttress thread classes available are 2 and 3, with tables given in ANSI B1.9 for each class.

These threads are made by milling or grinding, rather than on an automatic machine.

American (0.6P) buttress threads are shown in Figure 11.

THREADS

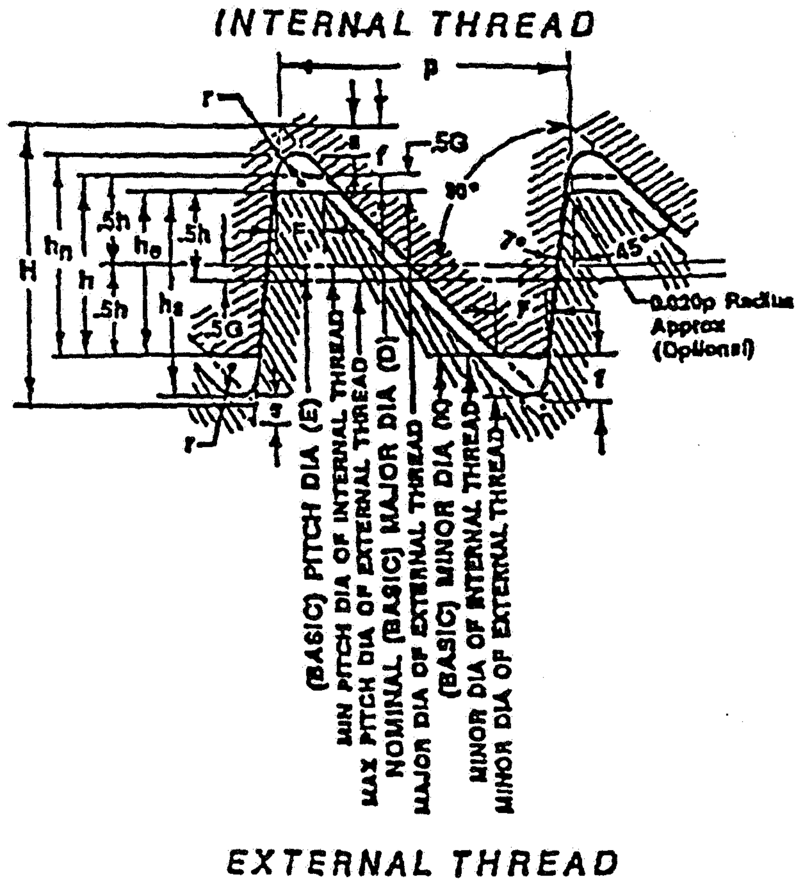


FIG. 1a FORM OF STANDARD 7°/45° BUTTRESS THREAD WITH 0.6p BASIC HEIGHT OF THREAD ENGAGEMENT AND ROUND ROOT

(Heavy line indicates basic form)

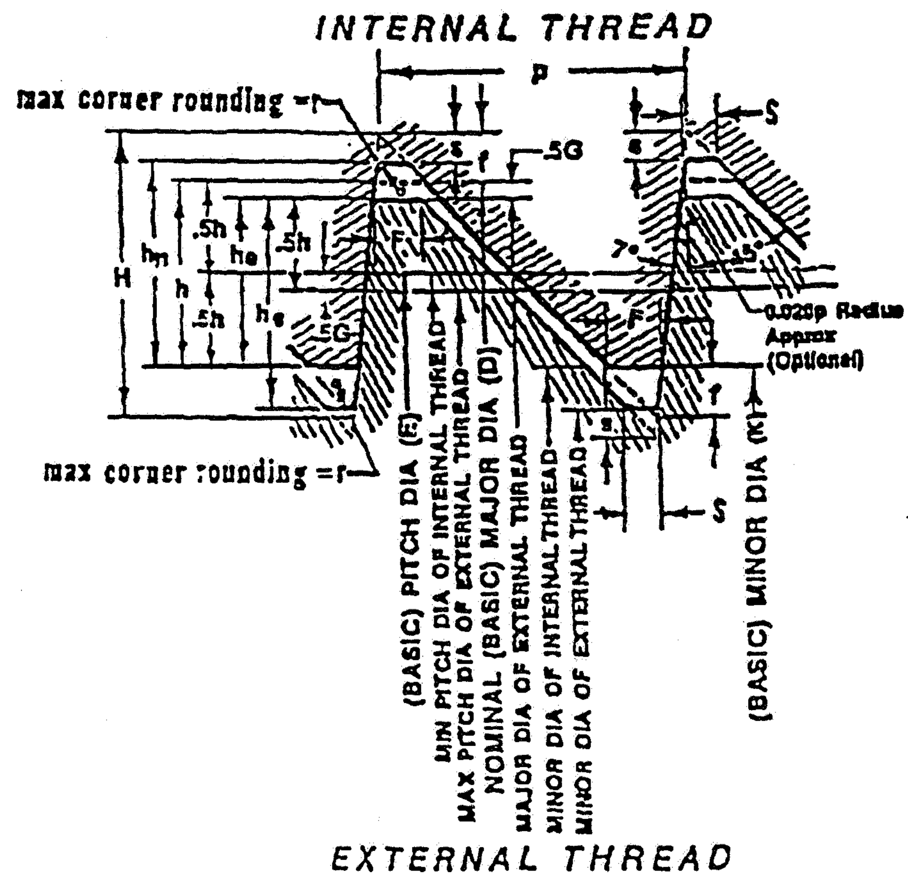


FIG. 1b FORM OF 7°/45° BUTTRESS THREAD WITH 0.6p BASIC HEIGHT OF THREAD ENGAGEMENT AND FLAT ROOT

(Heavy line indicates basic form)

Figure 11 - Buttress Threads

THREADS

- **Cross-Sectional Areas for Stress Calculations**

Different cross-sectional areas are used for tension and shear stress calculations. If the fastener is loaded in shear, with no threads in the shear plane of the hole, the full shank area can be used for shear stress calculations.

For tensile stress calculations, a minimum area through the threaded portion of the fastener is used but this area is not a circle with a diameter of twice the root radius. Since there is a thread crest diametrically opposite the root, the effective diameter is slightly larger.

THREADS

- **Cross-Sectional Areas for Stress Calculations (Cont'd)**

The formula for calculating the thread tensile area is:

$$A = \frac{\pi}{4}\left(D - \frac{0.9743}{n}\right)^2$$

where "n" is the threads per inch and D is the shank diameter (in).

For Metric fasteners:

$$A = \frac{\pi}{4}(D - 0.9382P)^2$$

where "P" is the thread pitch (mm) and D is the shank diameter (mm).

A derivation of the tension area formula is given in Appendix 1.

THREADS

- **Cross-Sectional Areas for Stress Calculations (Cont'd)**

For shearing off the threads, rather than failing the cross section in tension, an empirical formula can be used to calculate the allowable pullout load.

$$P = \frac{\pi D_m F_s L}{3}$$ *where:*

P = *allowable pullout load (lbs.)*

D_m = *mean diameter of threaded hole (approximate pitch diameter) (inches)*

L = *length of full thread engagement (inches)*

F_s = *material shear ultimate or shear yield allowable stress (psi) for the weaker of the two materials*

THREADS

- **Cross-Sectional Areas for Stress Calculations (Cont'd)**

The 1/3 factor is empirical to allow for thread mismatch, since it would be 1/2 if the threads were perfectly mated. The two cylindrical shells of material would then be the same for the internal and external mating threads.

This formula can be used for metric calculations, as long as the units are consistent.

An exact method of calculating this area is given in Appendix 2.

FATIGUE RESISTANT BOLTS

- Introduction

If a bolt is subjected to cyclic loading (fatigue), an attempt should be made to minimize the stress risers created during its manufacturing cycle. Some of these areas are the threads, thread runout, head fillet radius, and work hardening during forming of the bolts.

In addition, the bolt installation should be closely controlled to minimize cycling modes.

FATIGUE RESISTANT BOLTS

- **Cold Rolled Threads**

Cold rolled threads are used where high strength and fatigue resistance are desired. The cold rolling process leaves residual compressive stresses in the thread surfaces. These stresses give the surface more fatigue resistance. The cold rolling process also raises the material strength. Some fasteners (such as 220 ksi A286 CRES) require both cold rolled threads and cold working of the material to reach the desired strength levels. Remember that "J" threads are better than regular threads in fatigue.

- **Elongation Limits on Material**

Since fasteners have many areas with stress risers, it is desirable to use fastener materials with a minimum elongation of 10% to minimize failures from stress risers.

Fatigue Resistant Bolts

- **J-Threads**

J-threads are more fatigue resistant than regular threads, since the thread root radius is larger *AND* is inspected.

- **Countersunk Washers Under Heads**

Where high strength fasteners are used in fatigue, a washer with a countersunk hole should be used under the head of the bolt. This will prevent load concentration on the fillet radius under the bolt head.

- **Undercut Shank To Minor Diameter**

If a bolt is cycled in tension, it will normally break near the end of the threaded area, due to stress concentration. To avoid this stress concentration, the shank can be machined down to the minor diameter of the thread.

FATIGUE RESISTANT BOLTS

- **Hardness of Nut Less Than Bolt Hardness**

  Since the bolt load is initially reacted by one to three threads on the nut, it is better to have the nut deform first to distribute the load. A design rule-of-thumb is that the maximum hardness of the nut should not exceed the minimum hardness of the bolt.

- **Use Desirable Joint Loading Diagram**

  Use a high joint/fastener stiffness ratio (5 or higher) to minimize cyclic loads. (The methods of calculating joint stiffness will be covered later.)

FATIGUE RESISTANT BOLTS

- **Avoid Tapped Holes**

  Tapped holes are cut, rather than rolled. The root radii are not measured and inspection is usually difficult to do. Therefore, they could have undetected stress risers which could lead to early fatigue failures.

- **Use Many Small Diameter Bolts**

  The use of many small diameter bolts gives a more elastic system to resist dynamic loading cycles. More uniform tension loading of these bolts is possible than if fewer (but larger) bolts are used.

FATIGUE RESISTANT BOLTS

- **Consider Thermal Loading of Joint**

  If the bolt and joint materials are different, it may be necessary to modify the bolt torque to accommodate differential thermal expansion/contraction. The operating temperature of the joint will determine the loads to be used for fatigue calculations. The use of Belleville washers may be required to limit maximum loads.

- **Torque Fasteners Close to Yield Point**

  If enough testing is done to determine where the fastener yield point is, the fastener can be torqued to 90 to 95% of yield for a fatigue joint. The higher preload decreases the stress cycling of the joint, thereby increasing its fatigue capacity. (See Figures 12 and 13 for bolt external loading diagrams.)

FATIGUE RESISTANT BOLTS

- **Triangle OAB is the Same in all Four Diagrams of Figure 12. OA Represents Bolt Stiffness and AB Represents Joint Stiffness**

  - ➛ In (a) the externally applied load F_e (a) does not load the bolt to its yield point.

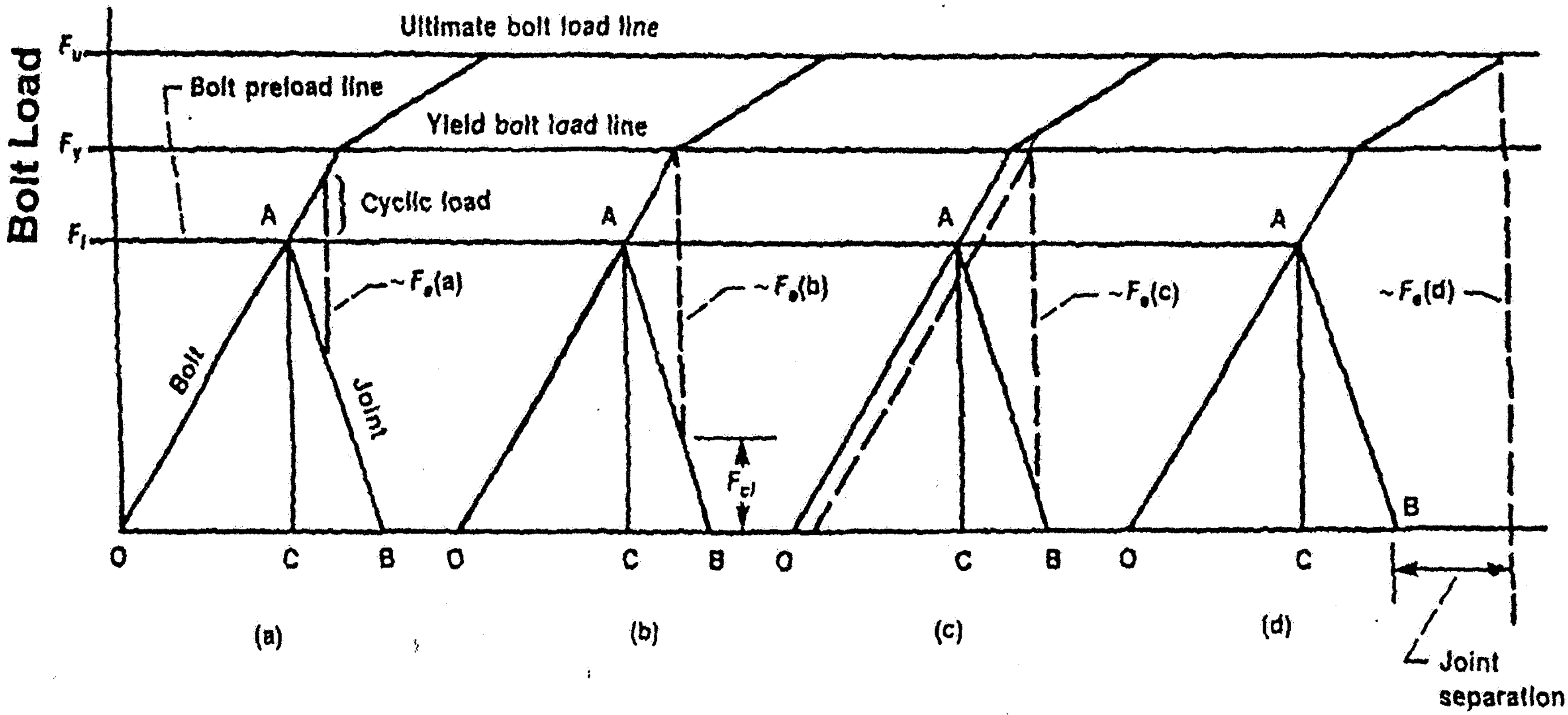
  - ➛ In (b) the bolt is loaded to its yield point by F_e (b).

  - ➛ In (c) the bolt is loaded above its yield point by F_e (c) and will have less than F_i load when F_e (c) is removed.

  - ➛ In (d) the joint has completely separated on its way to bolt failure.

FATIGUE RESISTANT BOLTS

Figure 12 - Bolt External Loading

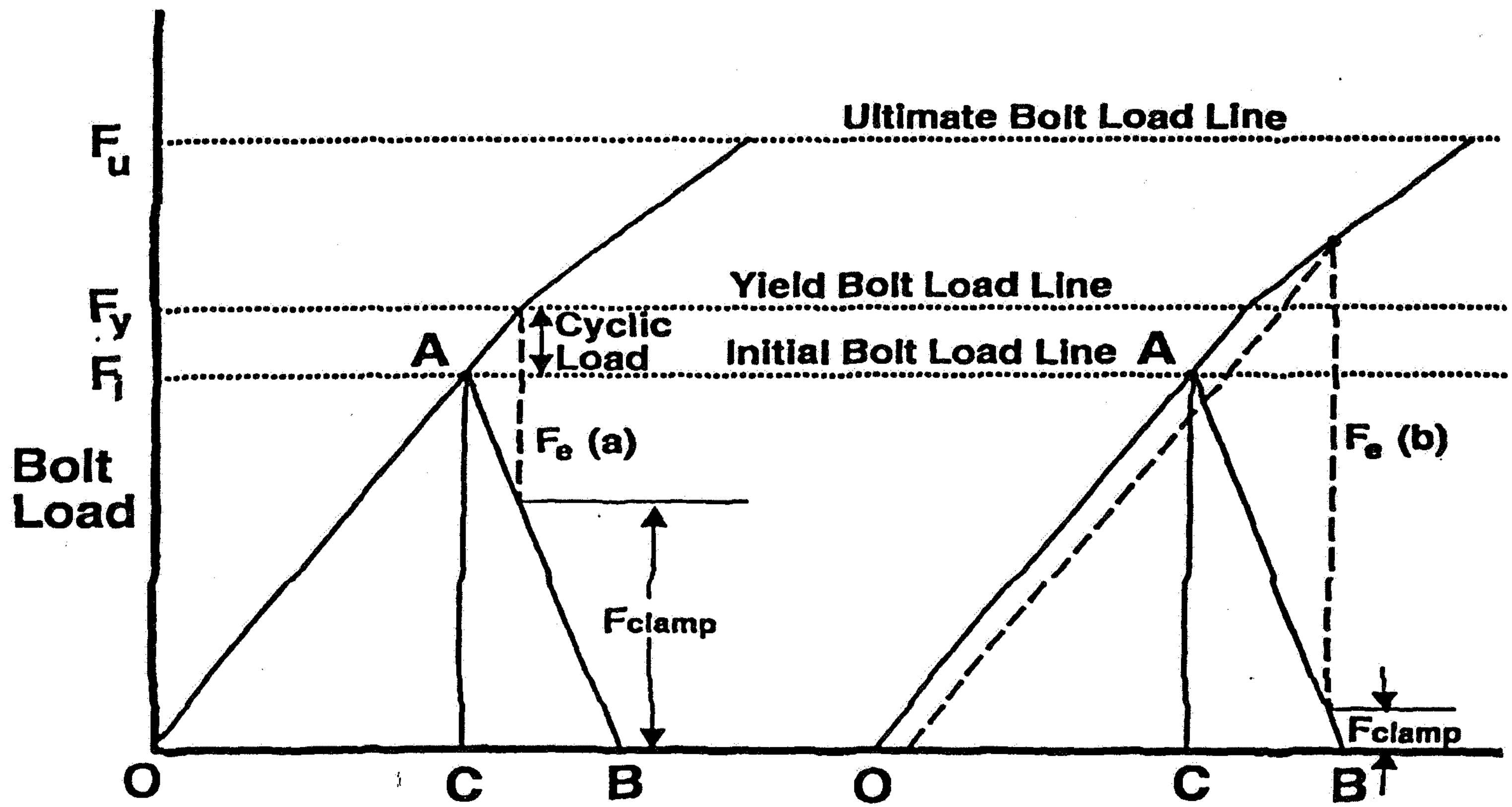


Elongation

FATIGUE RESISTANT BOLTS

Figure 13 - Bolt External Loading Diagram

(With Higher Stiffness Ratios and F_i Near Yield)



(a) Load to Yield

(b) Load above Yield

Elongation

FASTENER TORQUE

- ## Introduction

Determination of torque values for fasteners is one of the most difficult and controversial aspects of fastener design. Some of the many variables involved in the torque value are:

- ➛ Joint material strength(s)
- ➛ The coefficient of friction between mating threads
- ➛ The effect of friction between the bolt head (or nut) and its mating surface
- ➛ The effect of coatings and lubricants on the friction coefficients

FASTENER TORQUE

- **Introduction** (Cont'd)
  - ➛ The percentage of bolt tensile strength to be used for preload
  - ➛ What is the distribution of total torque to tension, shear, and friction?
  - ➛ Relative spring rates of the bolts and joint
  - ➛ Accounting for running torque of locking devices

FASTENER TORQUE

- **Head Friction**

If a fastener is tightened from the head, the bearing surface for the bottom of the head becomes critical for friction loads. A hardened smooth washer should be used under the head, along with a suitable lubricant, to minimize friction. The washer will also deter or prevent embedment of the head where the joint material is much softer than the bolt.

If head friction locking is desired, then maximizing head friction becomes the goal. Use of a serrated head face or a serrated washer without lubricant would be appropriate. However, the amount of torque required to overcome head friction must be accounted for in determining bolt tension.

FASTENER TORQUE

- **Nut Friction**

  The nut requires a hardened smooth washer also, as well as lubrication, to minimize friction. In addition, the nut usually contains the locking device. The running torque of the locking device must be accounted for in the total torque value.

FASTENER TORQUE

- ## The Elusive "K" Factor

A popular formula for quick torque calculations is T=KFd, where T is torque, F is axial load and "d" denotes bolt shank diameter. K (torque coefficient) is a calculated value using the formula:

$$K = \left(\frac{d_m}{2d}\right) \frac{\tan\lambda + \mu \sec\alpha}{1 - \mu \tan\lambda \sec\alpha} + 0.625\,\mu_c$$

as given in *MECHANICAL ENGINEERING DESIGN* (4th Ed.) by Shigley & Mitchell,

Where d_m = mean thread diameter
λ = thread lead angle
μ = friction coefficient between threads
α = thread angle (normally 30°)
μ_c = friction coefficient between bolt head (or nut) and clamping surface

FASTENER TORQUE

- **"K" Factor** (Cont'd)

The commonly assumed value for K is 0.2, but this value should not be used blindly.

Some calculated values for K are given in Table 10. I used equal values for μ and μ_c, but they are not necessarily equal. A more realistic "typical" value for K would be 0.15 for steel on steel.

Friction coefficient		Torque coefficient, K
Between threads, μ	Between bolthead (or nut) and clamping surface, μ_c	
0.05	0.05	0.074
.10	.10	.133
.15	.15	.189
.20	.20	.250

Table 10 - TORQUE COEFFICIENTS

FASTENER TORQUE

- **Torque Definitions (Courtesy of SAE AS1310 and MSFC Std 486)**

  - ➛ *TORQUE* - Force applied at the end of the length of a moment arm creating a turning moment (inch-pounds, foot pounds, or newton-meters).

  - ➛ *APPLIED TORQUE* - Torque transmitted to the fastener by the installation tool (torque wrench, impact wrench, or non-measuring wrench).

  - ➛ *RUNNING (OR PREVAILING) TORQUE* - Torque required to overcome static friction when 100 percent of the locking feature is engaged and the fastener is not seated. Running torque does not include torque required to increase or decrease the axial load in the fastener.

FASTENER TORQUE

- **Torque Definitions (Cont'd)**

  ➛ ***DOUBLE TORQUE (RETORQUE)*** **- Torque to seat materials being joined where moderate interferences, sheet gaps or form-in-place gasket material, are present in the assembly. It includes loosening the fasteners after the first application of installation torque and retorquing to ensure proper assembly.**

  ➛ ***FREE RUNNING TORQUE (NO LOAD TORQUE)*** **- Torque required to overcome kinetic friction between mating threads. It can be measured in either the loosening or tightening direction. Free running torque does not include torque required to overcome a self-locking feature or to increase or decrease the axial load in the fastener. Under normal conditions, free running torque is negligible.**

FASTENER TORQUE

- **Torque Definitions (Cont'd)**

  ➢ *INSTALLATION TORQUE (ASSEMBLY TORQUE, TIGHTENING TORQUE)*
  Prescribed design torque applied in the tightening direction at assembly and includes the net effect of:

  - **Torque required to overcome kinetic and static friction between mating bearing faces and mating threads.**

  - **Torque required to overcome any self-locking feature that may be present.**

  - **Torque required to apply desired axial load to the fastener assembly.**

  Installation torque is measured in the tightening direction only.

FASTENER TORQUE

- **Torque Definitions** (Cont'd)

  - *LIMITING TORQUE* - Predetermined torque level, which when reached during the torquing operation, results in break-away of a torquing element attached to a fastener or torque wrench slippage at the predetermined torque level.

  - *MULTIPLE TORQUE* - Torque required to seat parts where relatively heavy interferences exist in assembly. Installation torque is applied several times until the nut or bolt does not rotate during two successive installation torque applications.

  - *NET TORQUE (PRELOAD TORQUE)* - Torque available for axially loading a fastener during installation torque application after all frictional and self-locking mechanical forces have been overcome.

FASTENER TORQUE

- **Torque Definitions (Cont'd)**

  ➢ *SEATING TORQUE* - **Torque required to bring bearing faces (head and/or nut) into seated position.**

  ➢ *UNSEATING (BREAK LOOSE) TORQUE* - **Torque required to loosen the fastener from its installed position.**

FASTENER TORQUE

- **What Part Tension?**

The most unpredictable part of fastener installation is the percentage of total torque which goes into clamp load. The exact amount is not known unless instrumentation or extensive joint testing programs are used. In general, the clamp load represents 10% to 25% of the applied torque. The balance is used to overcome friction. However, this does not mean that the fastener has only a small load on it. The rest of the torque puts shear and torsional loads into the fastener. All loads must be combined and the stresses calculated to get a true state of fastener loading.

FASTENER TORQUE

- **What Part Tension? (Cont'd)**

  For fasteners, the von Mises stresses can be calculated and compared to the yield and ultimate material allowables. However, this is usually not practical, so calculated stress ratios are used to determine fastener margins of safety. (This topic will be covered later.)

NOTE: **Torque values for both inch and metric fasteners are given in the Appendix.**

FASTENER TORQUE

- ## Torque Accuracies

Torque accuracy is only as good as the type of measuring device and the operator. Of all methods in use, the impact wrench is probably the worst. If mechanics at garages would be honest, they would admit that their impact wrenches haven't been calibrated in years and that they are "good and tight," with no further accuracy claims. They usually use a "one torque fits all" setting. If a torque wrench is used to apply torque, the applied torque should be at least 70% of full scale. (This is a common guideline for most gage usage.)

Relative torque accuracies and costs are shown in Table 11.

FASTENER TORQUE

Table 11
Torque Measuring Methods vs. Accuracy and Cost

Preload Measuring Method	Accuracy, percent	Relative Cost
Feel (operator's judgment)	±35	1
Impact wrench	±35	1.5
Torque wrench	±25	1.5
Turn of the nut	±15	3
Load-indicating washers	±10	7
Fastener elongation	±3 to 5	15
Strain gages	±1	20
DTI Bolts	±2	25

FASTENER TORQUE

- Torque Striping

After final torquing of a critical bolt/nut assembly, a stripe of paint or ink is applied (usually on the nut end) across the end of the nut and bolt. Now if the nut or bolt moves after assembly, the movement can be detected by visual inspection.

NASA flight drawings carry a note to do the torque striping with a "Blue Sharpie" pen.

FASTENER TORQUE

- ## Joint Relaxation

First of all, the joint relaxation covered here is not "happy hour." It is defined as the unloading of a fastener after its final torque due to a number of contributing factors. Some of these factors are:

- ➣ Embedment of the washer, head, or nut in the joint material.
- ➣ Yielding of a high spot or blemish on the head, nut, washer or joint surface after final tightening.
- ➣ Untwisting of the fastener (from initial torsion) where the shank had an interference fit in the hole.
- ➣ Creep of the joint material.

FASTENER TORQUE

- **Joint Relaxation (Cont'd)**
  - ➛ Failure of the installer to retorque a pattern of fasteners after their initial installation to compensate for effects of adjacent fasteners to each other.
  - ➛ Inadvertently exceeding the yield point of the fastener during the initial torquing process.
  - ➛ Critical joints should be inspected for relaxation a few hours after installation.

FASTENER TORQUE

- **Turn of the Nut**

This is a method of tightening a nut to give a load above yield on a *DUCTILE BOLT*. It is accomplished by first tightening a bolt/nut assembly to approximately 75% of ultimate load. Then the nut position is marked. The nut is then turned an additional 180°. This brings the bolt stress level up above yield but below ultimate, providing that the material is ductile.

Turn-of-nut installations are used for larger diameter structural steel bolt assemblies. This method is *NOT USED* for aerospace installations. Aerospace torque values usually give tensile stresses of 50% to 75% of yield, depending on the application.

FASTENER TORQUE

- **Tightening Beyond Yield**

Tightening a fastener beyond its yield point is risky. Yield is difficult to determine without going above it. (This is why the yield strength of a material is defined as 0.2% permanent set.)

The usual reason for desiring near-yield conditions is to minimize fatigue effects on the fasteners. However, if a fastener is at the minimum strength allowed by the specification, it could yield enough to partially unload, thereby defeating the purpose of going to the high torque.

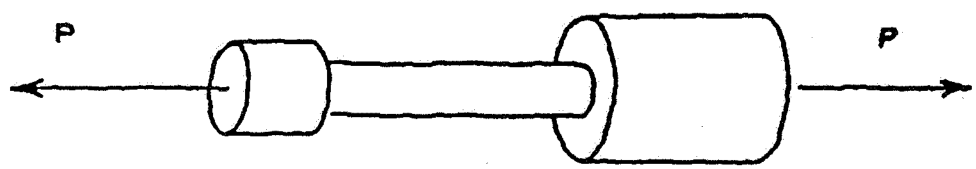
I personally do not recommend the practice of deliberately torquing a fastener beyond its yield point.

JOINT STIFFNESS

- ## Introduction

We have covered the joint loading diagrams in the bolt fatigue section, but we now need to look at the physical changes of the joint as we tighten the fasteners. The calculation of joint stiffness and fastener stiffness must also be looked at.

John Bickford[1] has used simple spring diagrams to illustrate joint and fastener stiffness. A fastener (rod) with variable cross sections is illustrated below.



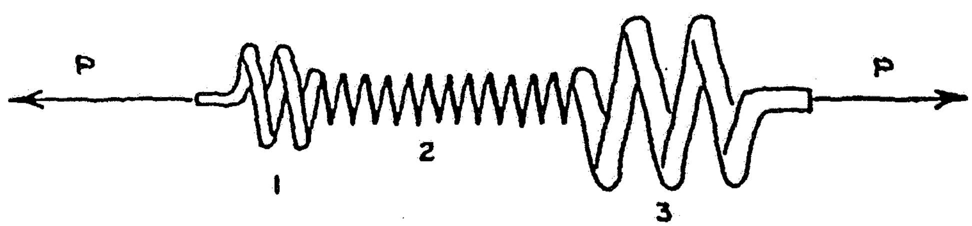
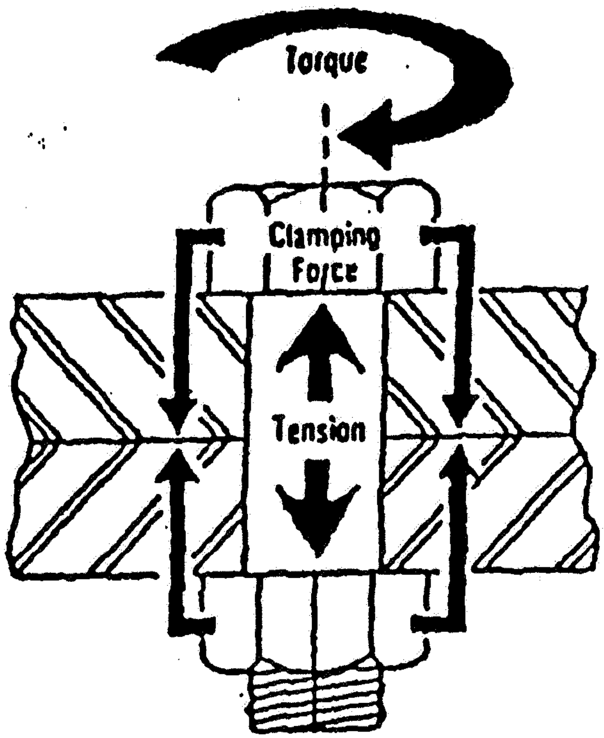


Figure 14 - Rod of Nonuniform Diameter, Loaded in Tension, and Equivalent Spring Model

[1] An Introduction To The Design and Behavior of Bolted Joints, 3rd Edition, J. Bickford, Marcel-Dekker Pub. Co.

JOINT STIFFNESS



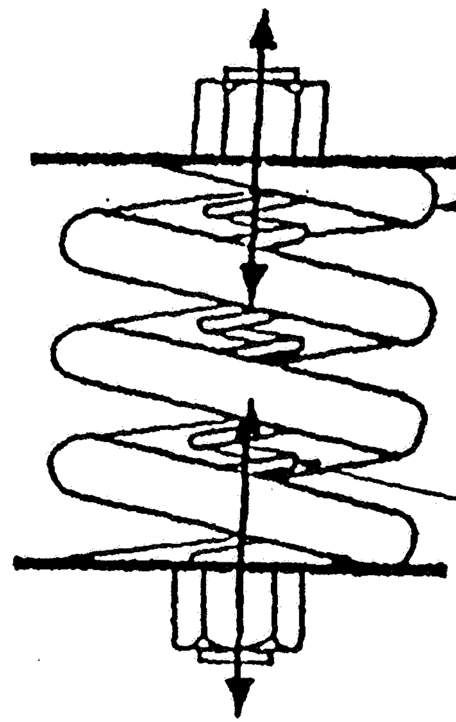


Figure 15
Bolt tension and joint compression due to bolt torque

Figure 16
Bolt stiffness and joint stiffness simulated with springs

JOINT STIFFNESS

- ## Bolt-Spring

Using Hooke's law for elastic deformation, the total change in the length of a rod with a uniform cross-section is

$$\Delta L = \frac{PL}{AE} \quad \textit{where:}$$

P = axial load (lbs.)
L = total elastic length of rod (in.)
A = rod cross-sectional area (in.^{2})
E = modulus of elasticity for the rod material (psi)

Applying this theory to a bolt, the total ΔL can be more accurately calculated by using ΔL_1, ΔL_2, ΔL_3, etc. for the different cross-sections and their lengths. An extreme case (from Bickford) is shown in Figure 17.

JOINT STIFFNESS

- **Bolt-spring** (Cont'd)

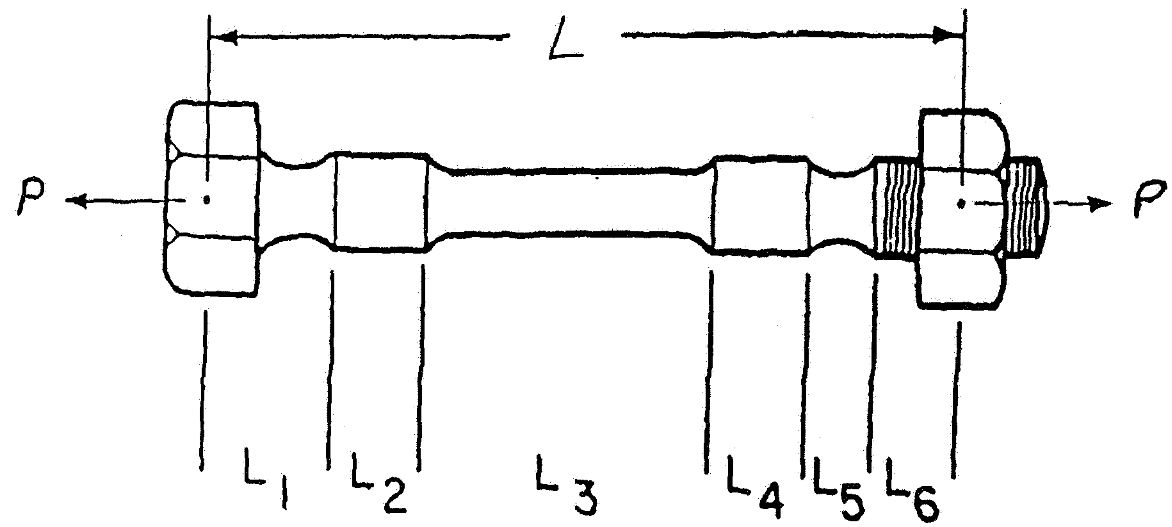


Figure 17 - Stiffness of a complex fastener

From the above figure:

$$\Delta L = \frac{P}{E}\left(\frac{L_1}{A_1} + \frac{L_2}{A_2} + \frac{L_3}{A_3} + \frac{L_4}{A_4} + \frac{L_5}{A_5} + \frac{L_6}{A_6}\right)$$

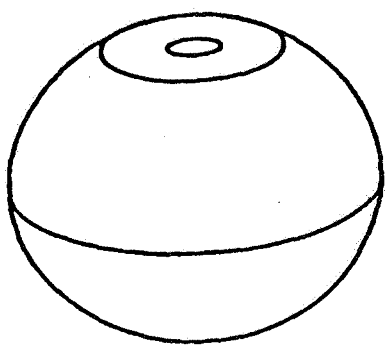
Note that L can be measured from the bottom of the bolt head if the head is a tension type. The length shown in the figure is from the midpoint of the head to the midpoint of the nut.

JOINT STIFFNESS

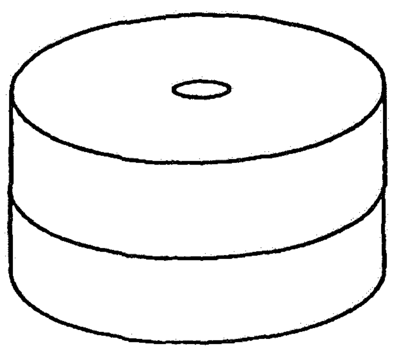
- **Clamped Material Stiffness**

  ➛ The joint stiffness is very similar to the fastener stiffness, except that the joint can be made of several different materials. In addition, the compressive effects of adjacent fasteners is not accounted for in the compression by an individual fastener.

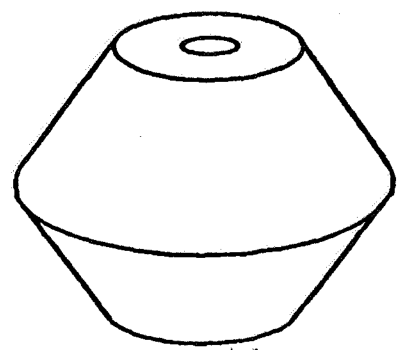
  ➛ Some empirical compression models considered by various authors are shown below (courtesy of J. Bickford).



(A) Sphere



(B) Cylinder



(C) Cone Frustum

Figure 18 - Empirical Shapes for Joint Stiffness Calculations

JOINT STIFFNESS

- **Clamped Material Stiffness (Cont'd)**

  - Bickford[1] uses the cylindrical model with a modification for eccentric loading at or near the edge of a joint. [This method is covered in the German Standard, Verein Duetscher Ingenieure (VDI).] [See Figure 18 (B)]

  - Shigley[2] uses the cone frustum model with a cone angle of 45°, measured from the bolt centerline. [See Figure 18 (C)]

  - NASA Langley[3] uses a straight cylinder with three different equations, depending on the minimum edge distance of the shortest side of the joint. [See Figure 18 (B)]

1 Introduction to the design and behavior of Bolted Joints, John Bickford, 3rd Edition, 1995, Marcel-Dekker

2 Mechanical Engineering Design, Shigley & Mitchell, 4th Edition, 1983, McGraw - Hill

3 NASA Langley SED Engineering Handbook EHB-2, 1990

JOINT STIFFNESS

- **Clamped Material Stiffness** (Cont'd)

  ➛ Blake[4] uses a cone frustum model with an angle determined by a line drawn from the outer edge of the head flat to the centerline of the clamped joint. This line extends to form a cone angle of 25° to 33° with the bolt centerline, with the exact angle being a judgement call or determined experimentally.

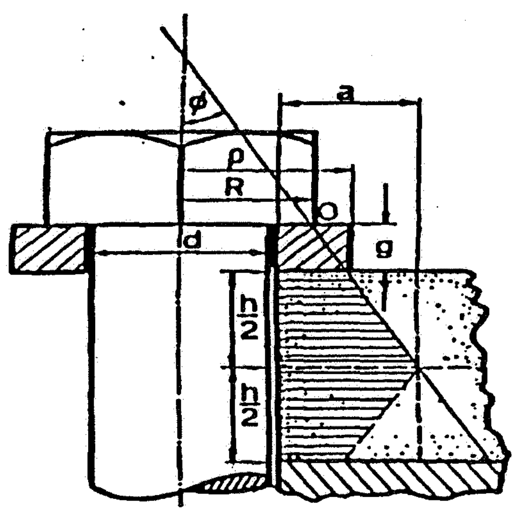


Figure 19 - Blake Cone Frustum Joint Area

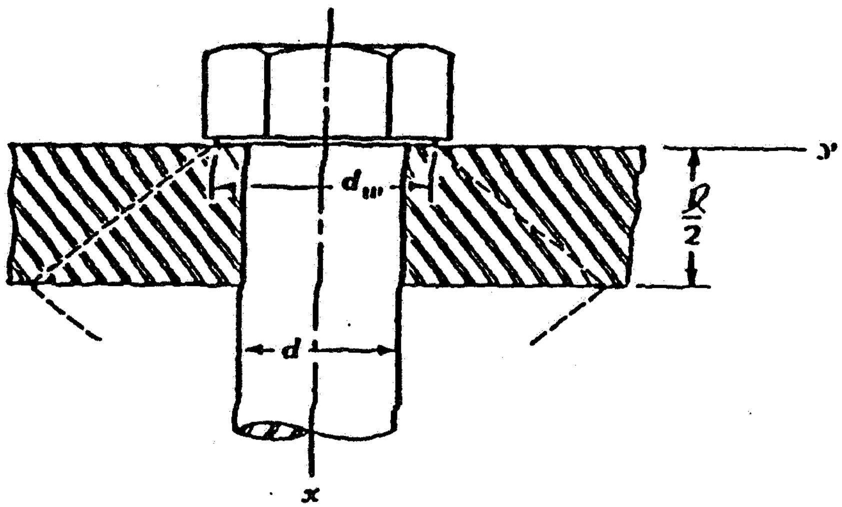
ϕ = cone angle
h = joint thickness
R = head radius to flat face
g = washer thickness
ρ = radius to washer O.D.
d = hole diameter (nominal)
a = a calculated value, which in this case is: $[a=\rho+0.5\,(h\tan\phi-d)]$

[4] Design of Mechanical Joints, A. Blake, 1985, Marcel-Dekker

JOINT STIFFNESS

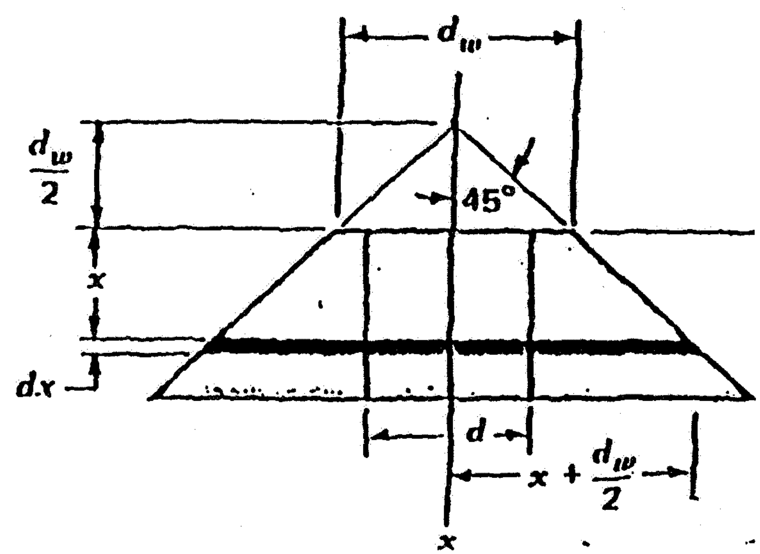
- **Clamped Material Stiffness** (Cont'd)

➔ **Shigley method for the frustum of a hollow cone. (See Appendix 3 for complete derivation.)**



Derived formula for joint stiffness is:

$$K_m = \frac{\pi \ Ed}{2 \ \ln \left[5 \left(\frac{\ell + 0.5d}{\ell + 2.5d} \right) \right]} = lbs/in.$$



Where:
E = Modulus of elasticity
d = Nominal bolt diameter
ℓ = Total thickness of joint members
ln = Natural logarithm
d_w = Contact area diameter
= Washer diameter (if washer is used).

JOINT STIFFNESS

- **Conclusions & Recommendations**

  - **Effect of adjacent fasteners on joint compression *IS NOT* accounted for in any of these methods.**

  - **Unsymmetrical loading under a fastener (due to edge distance or cutouts) *IS NOT* accounted for.**

  - **If bolt and joint materials are different, the stiffness calculations must account for the different moduli of elasticity. Temperature cycling must also be checked with the different expansion coefficients.**

JOINT STIFFNESS

- **Conclusions & Recommendations (Cont'd)**
  - ➢ First try a simple cylinder with a radius equal to the shortest edge distance on the fastener. If this stiffness is satisfactory, go no further.
  - ➢ If the simple cylinder is not satisfactory, add a washer with a diameter larger than the fastener head. Then use the Shigley cone frustum method to calculate joint stiffness.
  - ➢ Check the compressive stress under the head contact area to show that compressive yield (embedment) will not occur under maximum clamping load.

DIRECT READING OF FASTENER TENSION

- Introduction

This question is frequently asked, “How can I determine the exact tension I have on a fastener for a given torque?” A direct reading is possible but usually not economically feasible for production assemblies. The technology exists, but it is expensive and labor intensive. The usual compromise is to test fasteners under the closest actual installation conditions possible to establish the torque value to give a specified tensile load. Then that torque value will be used in production assemblies.

Some of the common tension measuring methods will now be covered.

DIRECT READING OF FASTENER TENSION

- Ultrasonic

  - ➛ A transducer is mounted to the head of the bolt. As the bolt elongates, the travel time for the sound wave increases. The elongation correlates directly with the bolt stress.

  - ➛ The major drawback to this method, other than cost increase, is that the transducer must have a smooth surface large enough for mounting. This usually requires grinding an area on the head. Most socket head bolts or recessed drive screws don't have enough smooth area to mount the transducer.

  - ➛ Another drawback to this method is that once the bolt is calibrated for zero load, the transducer usually must be removed during the torquing process.

DIRECT READING OF FASTENER TENSION

- ## Direct Scaling

In cases where both ends of the installed bolt are accessible (such as a pipe flange), the increase in bolt length after tightening can be measured with a vernier scale. However, the accuracy is only as good as the care exercised by the person making the measurement. The active length of the unloaded bolt must be used as a baseline for the elongation calculation.

- ## Direct Tension Indicating Washer (DTI)

These washers were covered in the washer section. To recap, they have protrusions which flatten at a calculated load. Then the gap between the protrusions and the mating surface is checked with a feeler gage to determine when the proper preload has been reached.

DIRECT READING OF FASTENER TENSION

- Test Machine by RS Technologies (Farmington Hills, Michigan)

  This test head will give a computer readout of the percentage of torque which goes to thread friction, head or nut friction, and direct tension for a given TEST BOLT. See Figure 20 for schematic.

DIRECT READING OF FASTENER TENSION

Fastener Research Test Head

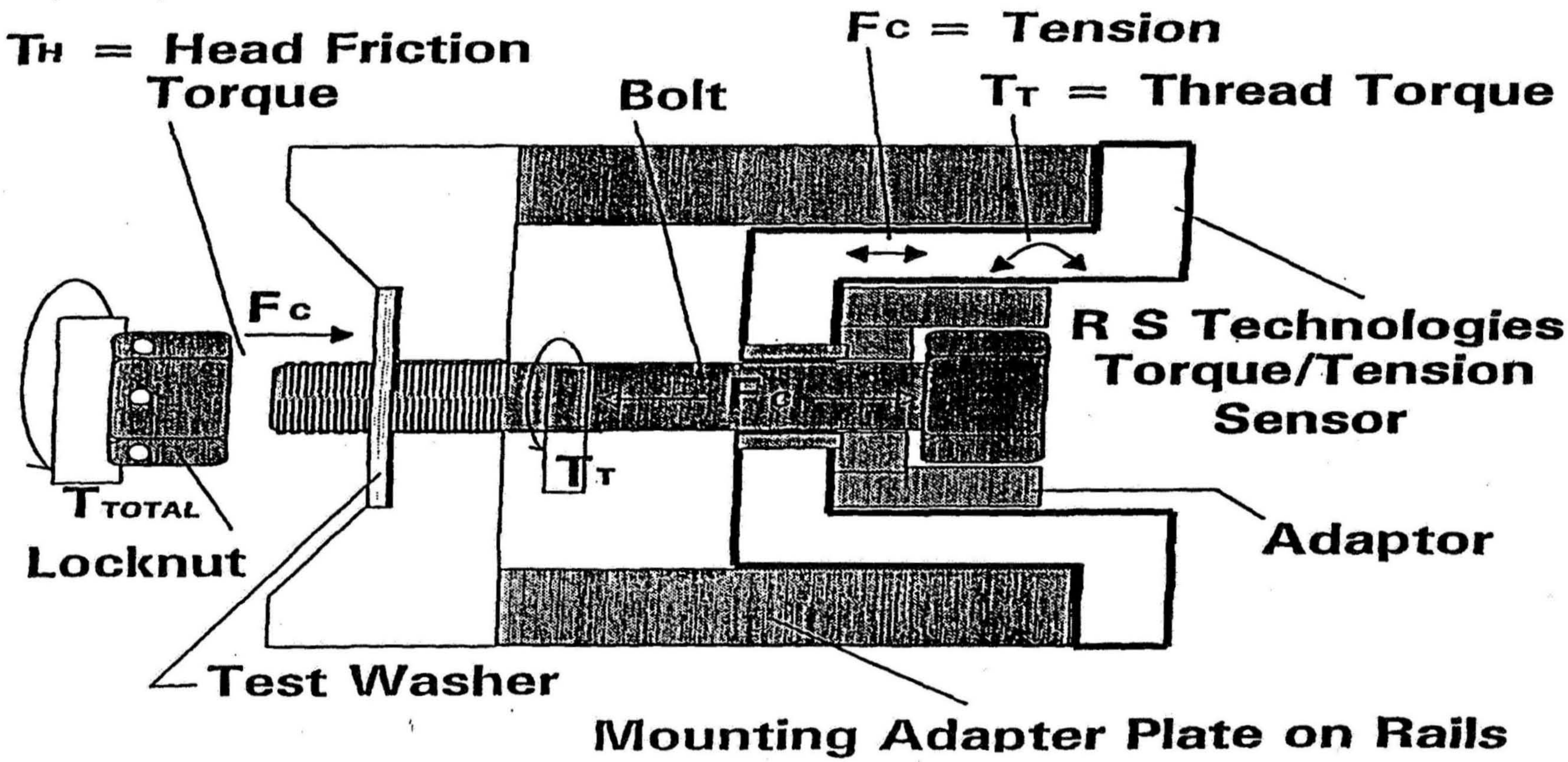


Figure 20 - RS Technologies Torque Measurement

DIRECT READING OF FASTENER TENSION

- Load Cells

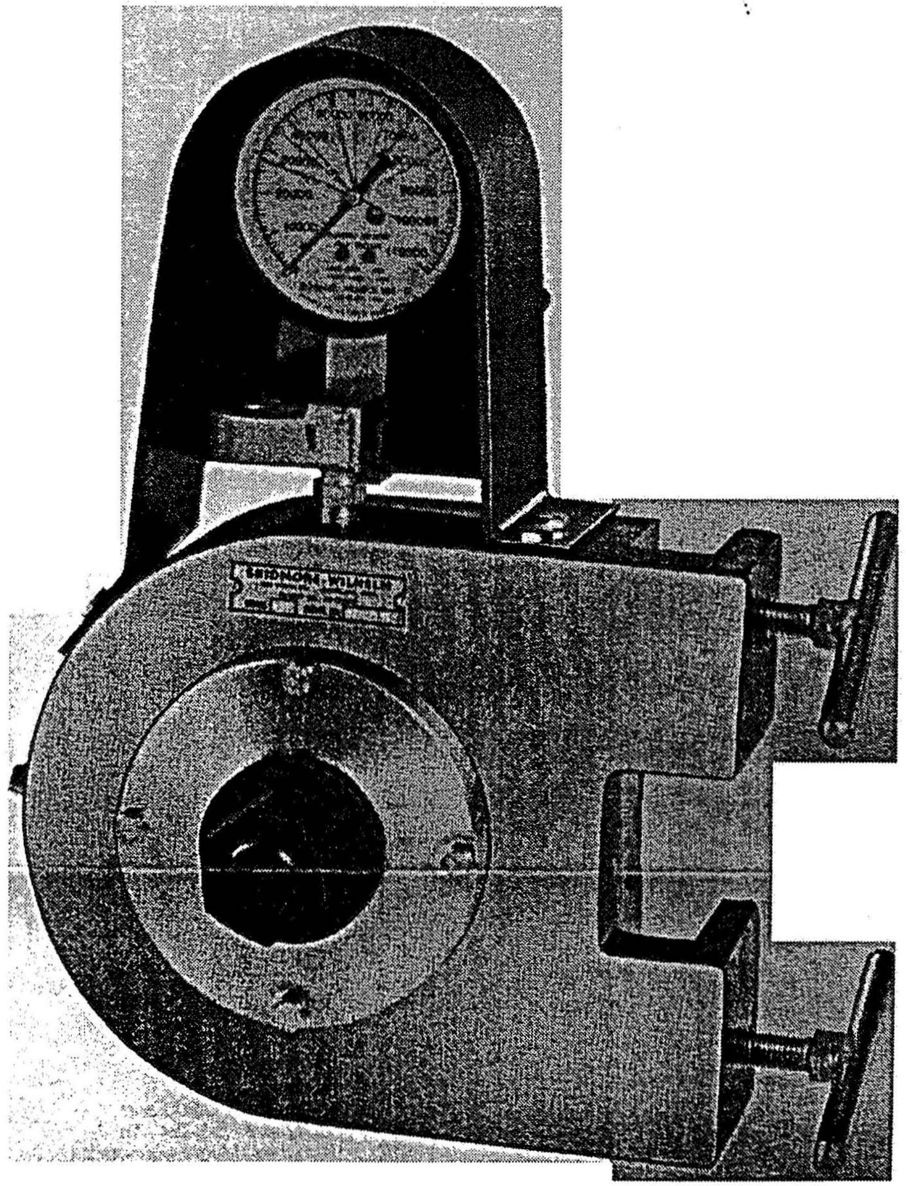
The Preload Indicating Washer (PLI), described in the washer section, is a mechanical load cell assembly. It is tightened until the outer ring won't turn. This amount of deformation has been correlated to a specific tension load in the bolt.

The Skidmore - Wilhelm bolt tension measuring machine, shown in Figure 21, has a load cell in it to give a direct bolt tension reading for an applied torque. It is a bench top device which can be clamped to a beam at a construction site for calibration of bolt samples before the entire lot is installed.

DIRECT READING OF FASTENER TENSION

Figure 21

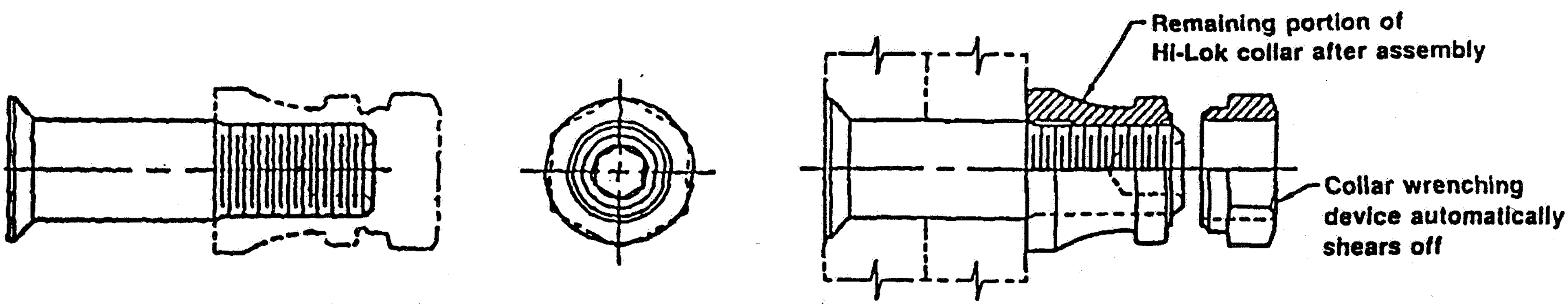
Skidmore - Wilhelm Bolt Tension Measuring Machine



DIRECT READING OF FASTENER TENSION

• Hi-LoK* Threaded Fastener

The Hi-LoK lockbolt has either a countersunk or a protruding manufactured head and threads like a bolt. It is installed from one side (blind). The installation gun prevents shank rotation with a hexagonal key while the nut is installed. The nut is notched to break off at the desired torque (as shown in Figure 22).



(A) Hi-Lok Pin

(B) Hi-Lok Pin and Collar After Assembly

Figure 22 - Hi-Lok Installation

*Hi-Shear Corporation, Torrance, California

DIRECT READING OF FASTENER TENSION

- ## Direct Tension Indicating (DTI™) Bolt

This bolt has an internal threaded gage pin with a head which fits against an optical absorptance cell near the bolt head surface. As the cell changes thickness, it changes color, giving a color - coded load indication, as shown in Figure 23.

Note that the minimum diameter for this type of bolt is .50 inch, and that these bolts are expensive.

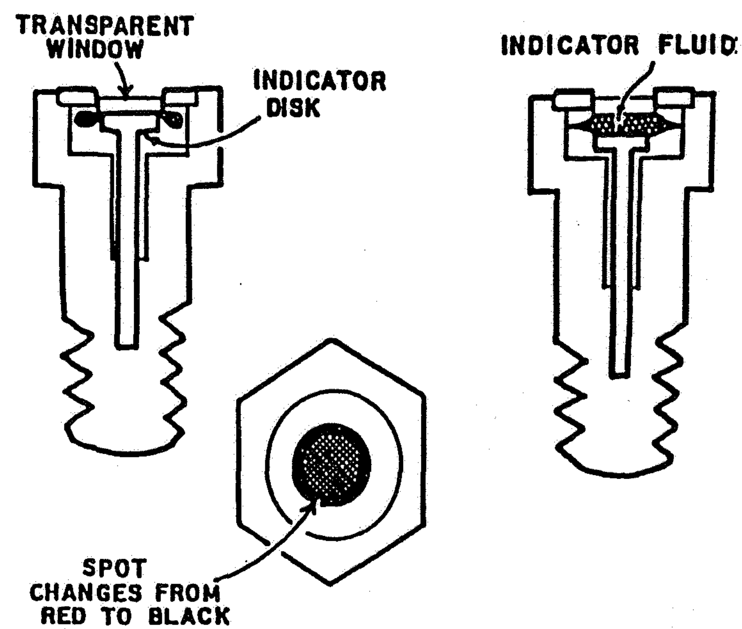


Figure 23 - ™Stress Indicators, Inc., Gaithersburg, MD

DESIGN CRITERIA

- Introduction

Sometimes we start a design before we think about it enough. We also do designs by an iterative process, where all of the requirements are not defined at the beginning. It's best to sit down and look at the "big picture" before starting on the design. Then see how many components can be bought from stock and how many must be custom designed.

Now look at accepted design practices, from both the layout and analytical standpoints. This section gives some of the things to look for, as well as some methods of analysis.

DESIGN CRITERIA

- ## Diameter Versus Length

  In designing a component assembly, we are always faced with decisions on using off-the-shelf or custom fasteners. A good preliminary procedure is to first see what is available and determine whether your design can incorporate standard stock sizes and materials as much as possible. Guidelines for lengths and diameters are given below.

  ➣ Length to Diameter Ratio - A L/D ratio up to 12 is usually available. (If 6 inch x .25 diameter fasteners are needed, they will be a custom item.) The L/D ratio is also limited by automated screw forming machine capacity.

  ➣ A listing of commonly stocked industrial fastener lengths versus diameters is given in Table 12.

DESIGN CRITERIA

Table 12
Hexagon-Head Machine Bolts-Stock Sizes (See Footnote)

Footnote:

One asterisk (*) represents stock sizes of maximum demand.

Two asterisks (**) represents stock sizes less frequently used.

All other sizes are considered specials.

Lgths. Inches	1/4" Dia.	5/16" Dia.	3/8" Dia.	1/2" Dia.	5/8" Dia.	3/4" Dia.	7/8" Dia.
1/2	**						
3/4	*	*	*	**			
1	*	*	*	*			
1-1/4	*	*	*	*	**	**	
1-1/2	*	*	*	*	*	*	
1-3/4	**	**	*	*	*	*	
2	*	**	*	*	*	*	**
2-1/4				**	**	**	
2-1/2	**	**	*	*	**	*	**
2-3/4				**			
3		**	**	**	*	**	**
3-1/2			**	**	*	**	**
4			**	**	**	*	**
4-1/2				**	**	**	**
5				**	*	*	**
5-1/2				**	**	**	
6				**	**	**	**

DESIGN CRITERIA

- **Calculation of "Number" Fastener Diameter**

  ➔ To get the diameter in inches use the following formula:

  Diameter = 0.060 + 0.013N where:

  N is the fastener number

  Example:

  A No. 8 fastener diameter is:

  D= 0.060 + 0.013(8) = .164 in.

DESIGN CRITERIA

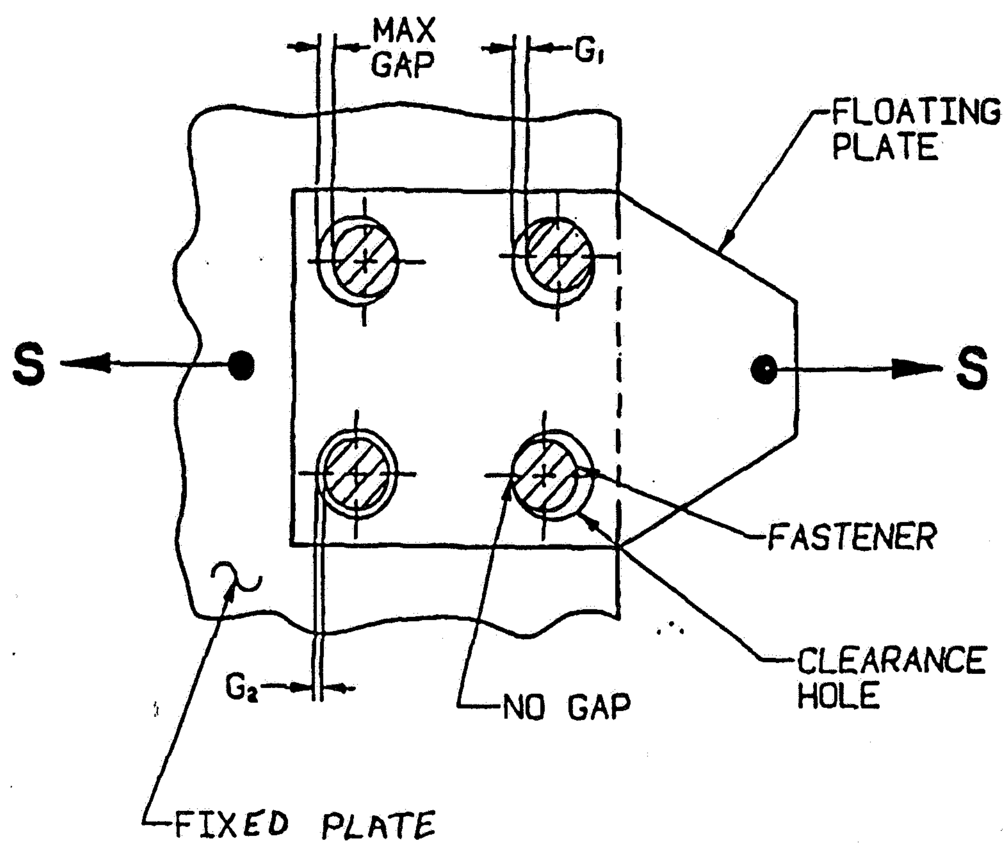
- **Clearance Holes for Fasteners**

➢ *SHEAR APPLICATIONS* - When the dominant load is shear, the clearance on the fasteners should be minimized. Ideally, the holes should be match drilled for the fasteners. The material thickness and fastener strength should be sized to make the fasteners critical in bearing rather than shear. An illustration of fastener clearance hole gaps is given in Figure 24.

➢ *TENSION APPLICATIONS* - If the tension loads are dominant to the point that the joint members do not slip laterally, the fastener clearance holes can have a looser fit. Then the main concern is to center the fastener head (and nut) with a washer to prevent embedment (compressive yielding) of the joint surface.

DESIGN CRITERIA

Figure 24
Clearance Hole Gaps on a Fastener Pattern (Before Loading)



DESIGN CRITERIA

- Mixing of Thread (and Material) Types

Mixing of different threads with the same diameter fasteners on the same assembly can be a disaster. If different sizes have fine, coarse (or metric) threads such that they only fit in the proper holes, this can be tolerated.

Mixing of fasteners of different materials or even the same material but at a different strength levels (e.g. Grades 2 and 8 steel fasteners) should not be done unless the fasteners *LOOK* very different. A mechanic sees fasteners that look alike as the same when they are laid out on a work bench. One of the other aerospace mistakes is to use 300 series and A286 stainless steel fasteners on the same component. The 300 series stainless has a yield of 30ksi and the A286 has a yield of 100ksi. Yet they look the same.

DESIGN CRITERIA

- Selection and Positioning of Washer

Although it seems mundane, it is important to pick washers that are large enough to distribute the head and nut loads without exceeding the compressive yield strength of the joint material. The washers should be smooth and harder than the fastener in order to minimize friction drag. If the internal diameter of the washer is much larger than the fastener diameter, care should be exercised to keep the washer centered while tightening the fastener.

DESIGN CRITERIA

- ## Shear Loads on a Fastener Group (Using Figures 25 and 26)

For analyzing a pattern of fasteners, the first task is to find the centroid of the pattern. This is done by the standard statics method of arbitrarily picking X and Y axes and using a unit area times its distance to get the X and Y of the centroid. Although it is poor design practice to have different sizes of fasteners in a pattern, they can be analyzed by picking the most common one as a unit value. Then ratio the other areas to this one. For example, if 8 bolts of a 12 bolt pattern are .375 dia. and the other 4 are .313 dia., the .375 dia. shank area of .1104 in.^{2} would be the unit value of one and the value for the .313 diameter fasteners would be .0767/.1104 or .69. The shank areas can be used for shear, since it is not good design practice to have threads in bearing. However, for tension loads, the thread root area should be used for calculations.

DESIGN CRITERIA

- **Shear Loads on a Fastener Group (cont'd)**

In many cases, the fastener pattern will be symmetrical, as shown in Figure 25, such that the centroid calculations are unnecessary. With the centroid located, the next step is to divide the load (R) by the number of fasteners (n) to get the direct shear load (Pc) on each fastener.
The next step is to find $\sum r^2_n$ for the pattern of fasteners. The r_n's are the radial distances from the centroid to the center of each fastener. In my symmetrical example, there would be 4 r_n's for each of only 2 radii, so that the $\sum r_n^2$ would be $4(r_1^2)+4(r_2^2)$. Now calculate the moment of the load R about the centroid (M=Re) from Figure 26. The shear load (lbs.) For a particular fastener due to this moment is:

$$P_e = \frac{Mr}{\sum r_n^2}$$

where r is the distance (in inches) from the centroid to the fastener in question (usually the outermost one).

DESIGN CRITERIA

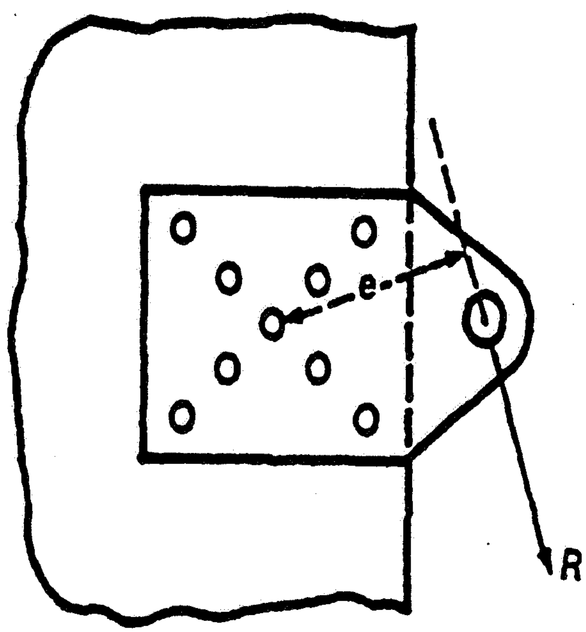


Figure 25
Symmetrical Load Pattern
(With Eccentric Load)

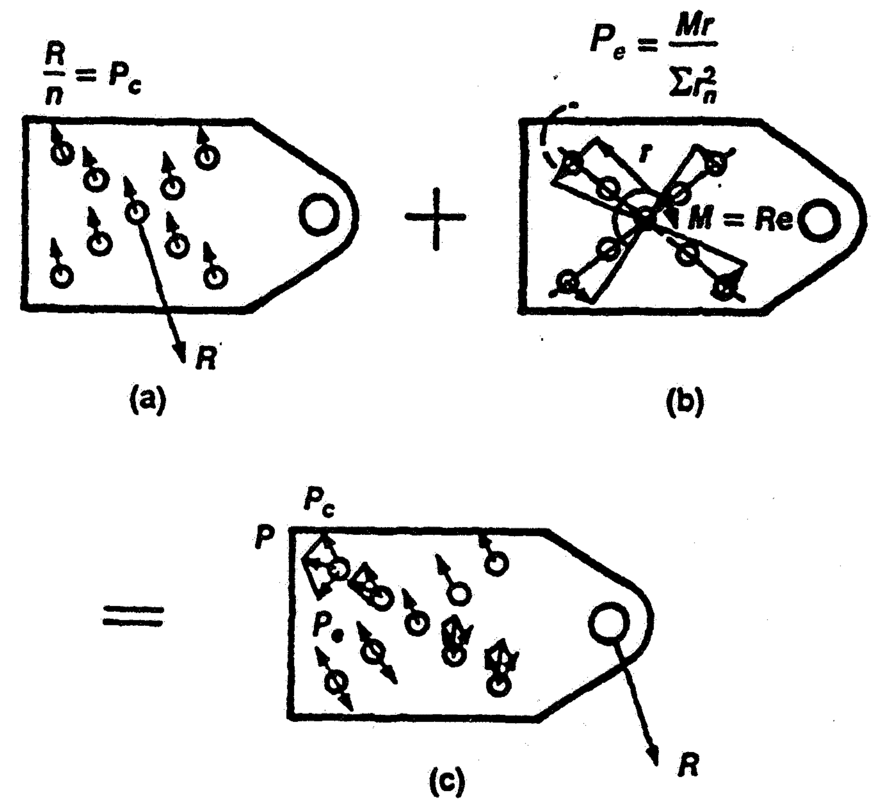


Figure 26
Combining of Shear and Moment
Loading

DESIGN CRITERIA

- **Shear Loads on a Fastener Group (Cont'd)**

Note that the expression for P_e is analogous to the torsion formula:

$$f = \frac{Tr}{J}$$

except that P_e is in pounds instead of stress. The two shear loads (P_e and P_c) can now be added vectorially to give the resultant shear load. This load must now be combined with the fastener tension load and compared to the total strength of the fastener, using load or stress ratios. (To be covered later in this course.)

DESIGN CRITERIA

- **Edge Distance & Fastener Spacing**

Although the actual edge distance and distance between fasteners can vary, a nominal value for edge distance is 2D with an absolute minimum of 1.5D. (Note that some aerospace companies use a nominal edge distance of 2D + .030 in.) Spacing is 4D. These distances are measured from the hole centerlines as shown in Figure 27. (Note that shear lugs are custom designs.)

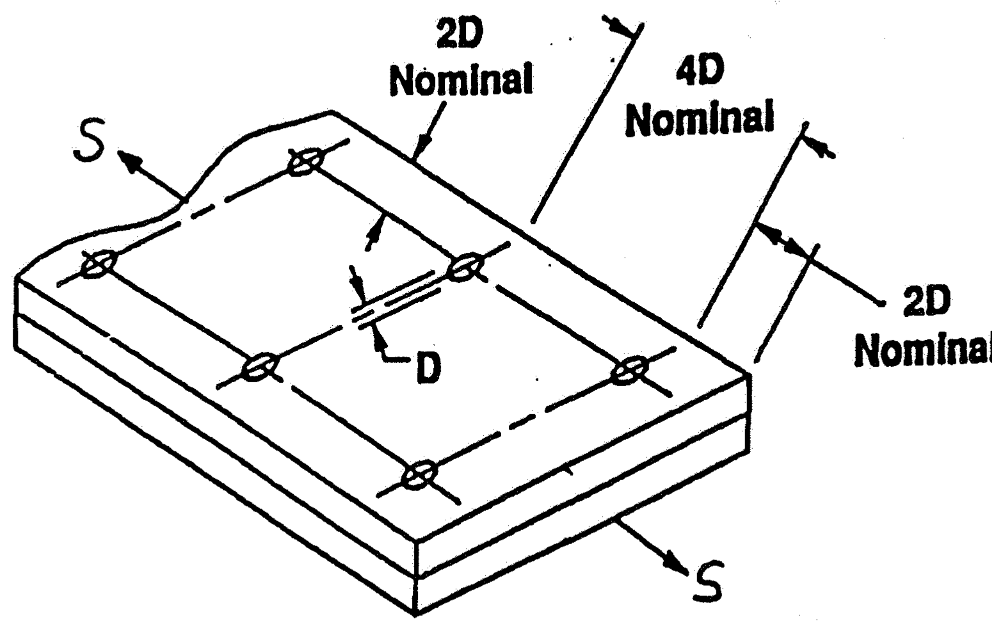
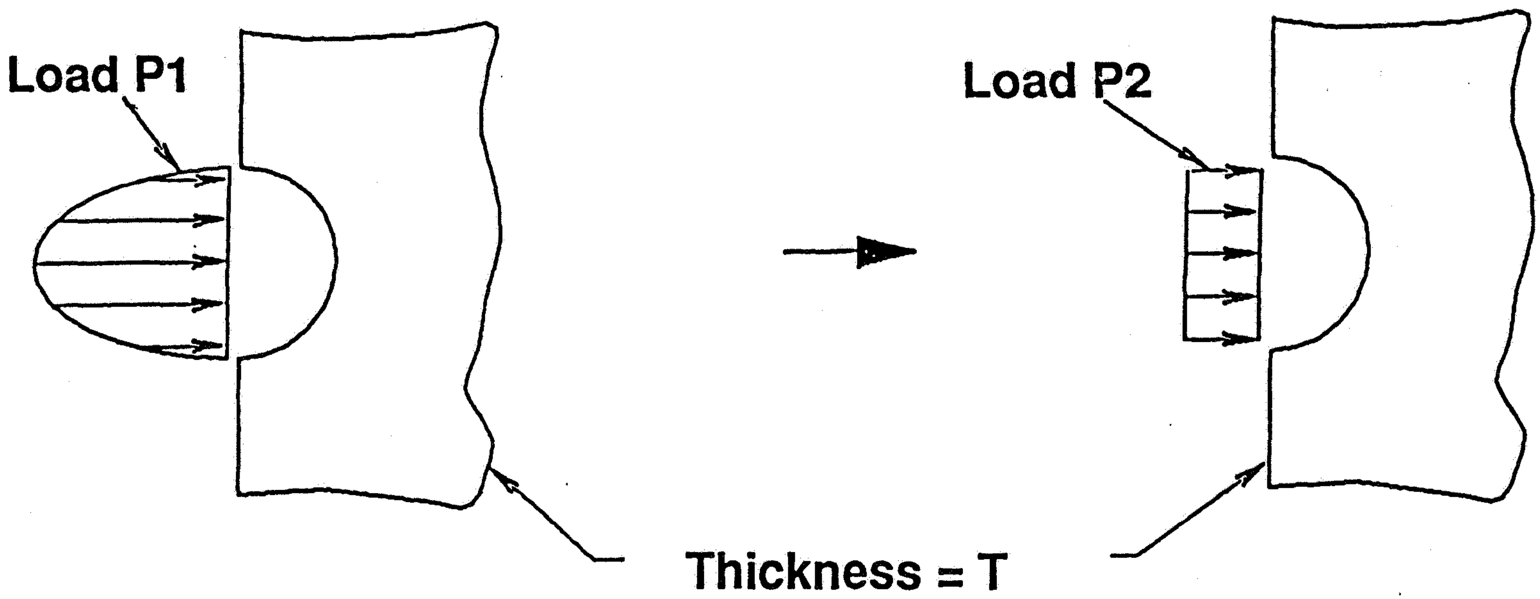


Figure 27 - Fastener Edge Distance and Spacing

DESIGN CRITERIA

- **Development of Bearing Stress Allowables**



$$\frac{P1}{F_{CY}} = \frac{P2}{F_{BRY}}$$

$$F_{BRY} \approx 1.5\, F_{CY}$$
$$F_{BRU} \approx 1.5\, F_{CU}$$

DESIGN CRITERIA

- **Grip Length, Shear Head, and Tension Head**

  For critical shear designs, it is mandatory to size the grip length of the threaded fastener such that there are no threads in the bearing area. This can be done by having a washer under the nut to allow tightening without running out of threads.

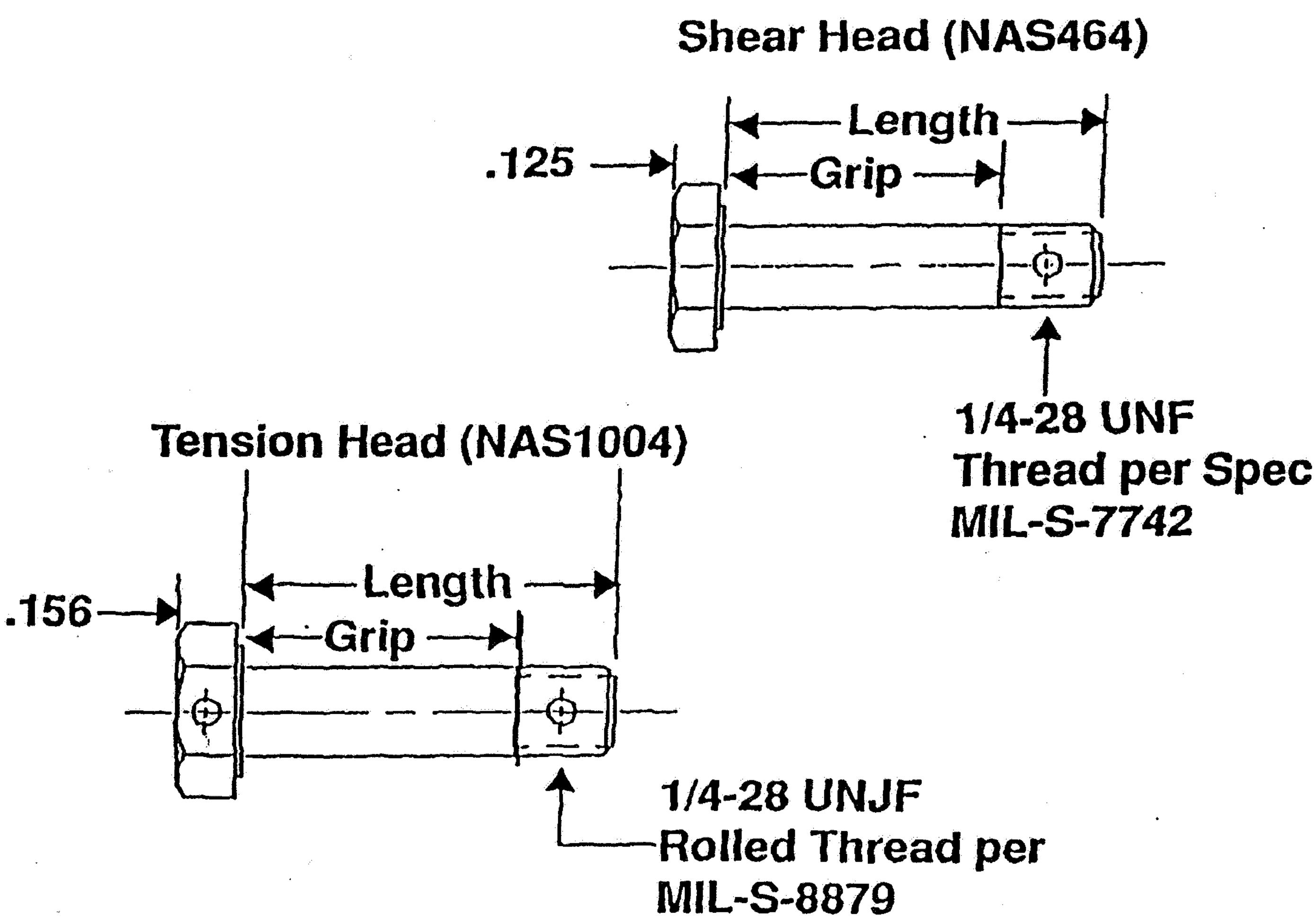
  Aerospace fasteners are available with shear heads/nuts or tension heads/nuts. This enables the designer to save weight in the areas where shear is the dominant load.

  Illustrations of grip length, shear head, and tension head are shown in Figure 28.

  A comparison of shear vs. tension nuts is shown in Figure 29.

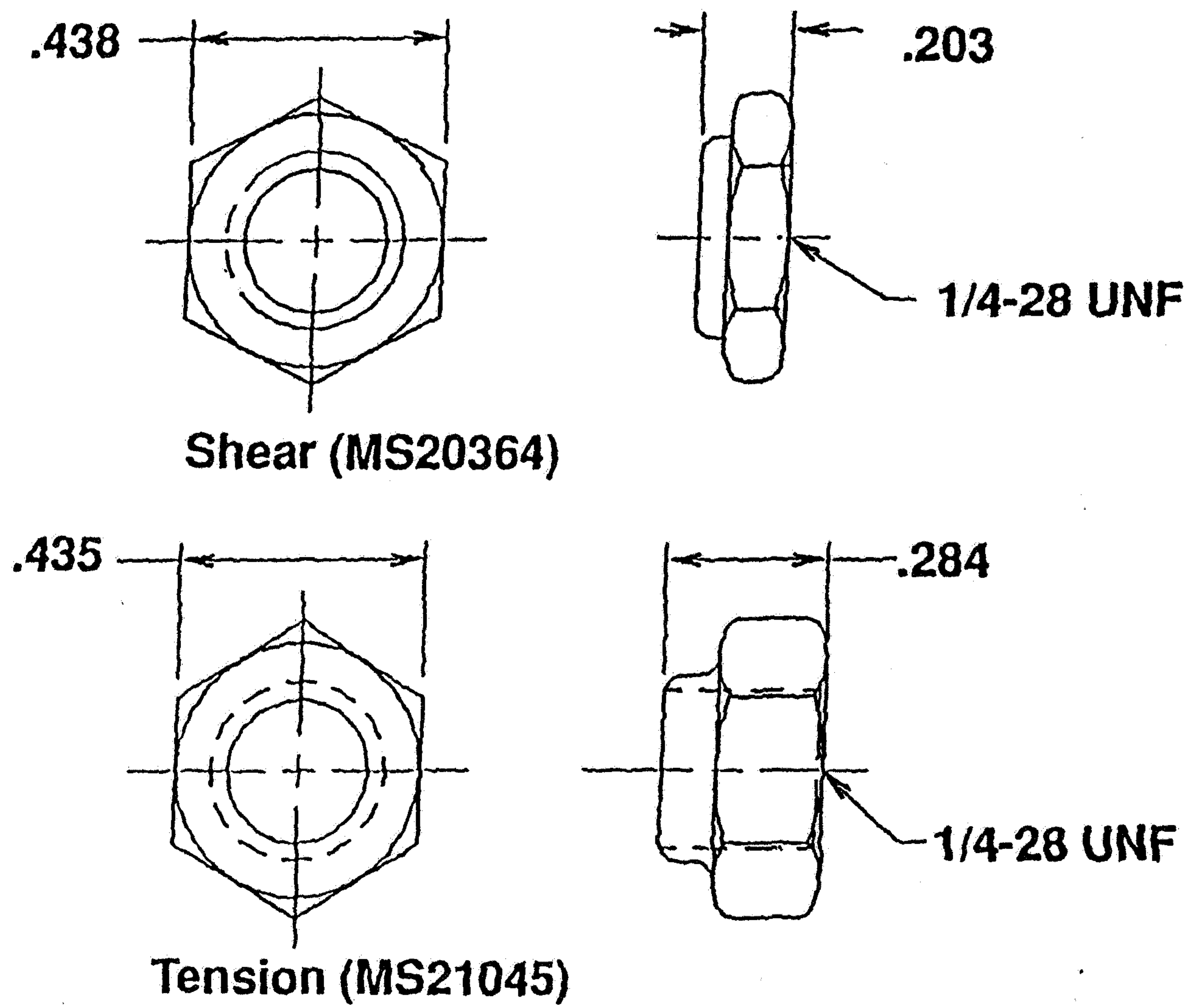
DESIGN CRITERIA

Figure 28 - Grip Length, Shear Head and Tension Head



DESIGN CRITERIA

Figure 29 - Shear Nut vs. Tension Nut



DESIGN CRITERIA

- ## Avoid Tapped Holes

Although tapped holes and the various types of taps were covered in the thread section, we should still avoid tapped holes as much as possible. Some important reasons for avoiding tapped holes are:

➛ Cost - Proper drilling and tapping of a hole is expensive, compared to drilling a clearance hole for a bolt and nut assembly.

➛ Inspection - About the only means of inspection for a tapped hole is a "Go-No-Go" gage and a minimum thread diameter check. (The root radius can't be successfully measured.)

➛ Quality - There is no such thing as a UNJ tap, so the root radius is not rounded. If the hole is blind, it will probably have some undetected burrs and shavings in it.

DESIGN CRITERIA

- ## Tension Loads on a Fastener Group

This procedure is similar to shear load determination, except that the centroid of the fastener group may not be the geometric centroid. It is easiest to illustrate with an example problem, using the bolted bracket of Figure 30.

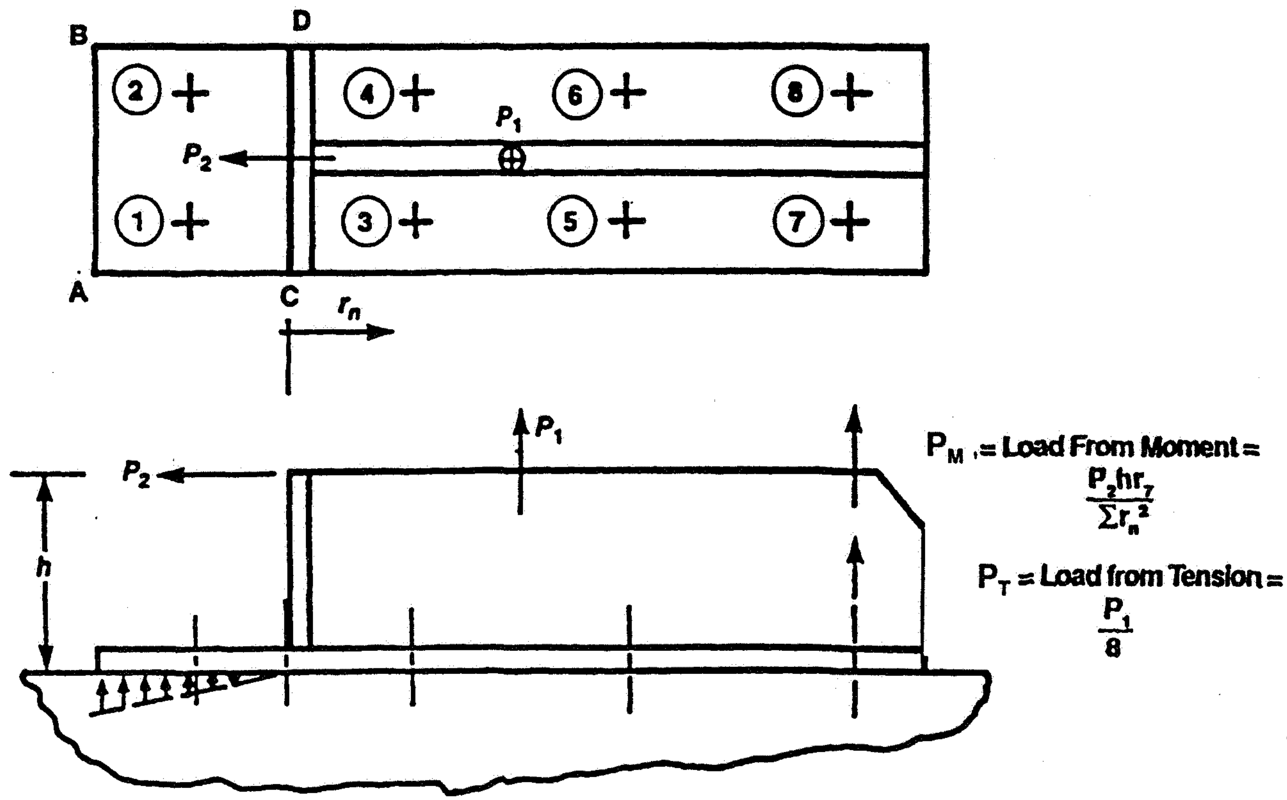


Figure 30 - Bolted Bracket with Shear and Tension Loading

DESIGN CRITERIA

- **Tension Loads on a Fastener Group (Cont'd)**

This pattern of eight fasteners is symmetrical about P_1, so $P_T = P_1/8$. The additional load of P_2 produces a shear load of $P_2/8$ per fastener and a moment of P_2h, which must be reacted by additional tension loads on some fasteners.

The problem is to determine where the bracket will go from tension to compression. This line is the "neutral axis" from which the "r" values are measured for the tension $\sum r_n^2$ determination. If the plate is thick enough to carry the entire moment (P_2h) at the edge (AB), that line could be used as the heeling point (or neutral axis).

DESIGN CRITERIA

- **Tension Loads on a Fastener Group (Cont'd)**

For this case, I have assumed that the plate is not strong enough to carry the moment and will go from tension to compression at line CD. Now the $\sum r_n^2$ will only include fasteners 3 through 8 and the r_n's will be measured from line CD (as shown).

Fasteners 7 and 8 will have the highest tensile loads (in pounds), which will be $P = P_T + P_m$, where:

$$P_M = \frac{Mr}{\sum r_n^2} = P_2 \frac{hr_7}{\sum Mr_n^2}$$

DESIGN CRITERIA

- **Tension Loads on a Fastener Group (Cont'd)**

The tensile preload for these fasteners must exceed P to prevent joint loosening.

To get the total fastener loads, the shear load, $P_2/8$, must be combined with P by stress ratios. Then the margin of safety can be calculated.

DESIGN CRITERIA

- ## Combined Tension and Shear Loading

When a fastener is simultaneously subjected to tensile and shear loading, the combined load must be compared to the total strength of the fastener.

Although a Mohr's circle could be used, the common procedure is to use stress or load ratios to calculate a margin of safety (MS)*. The load ratios are:

$$R_S (\text{or } R_1) = \frac{\textit{Actual Shear Load}}{\textit{Allowable Shear Load}}$$

$$R_T (\text{or } R_2) = \frac{\textit{Actual Tension Load}}{\textit{Allowable Tension Load}}$$

$$\text{*Margin of Safety} = \frac{\textit{Allowable Load (or Stress)}}{\textit{Actual Load (or Stress)}} - 1$$

DESIGN CRITERIA

- **Combined Tension and Shear Loading**

For our load ratios, the M.S. = $\dfrac{1}{R_s^x + R_T^y} - 1$

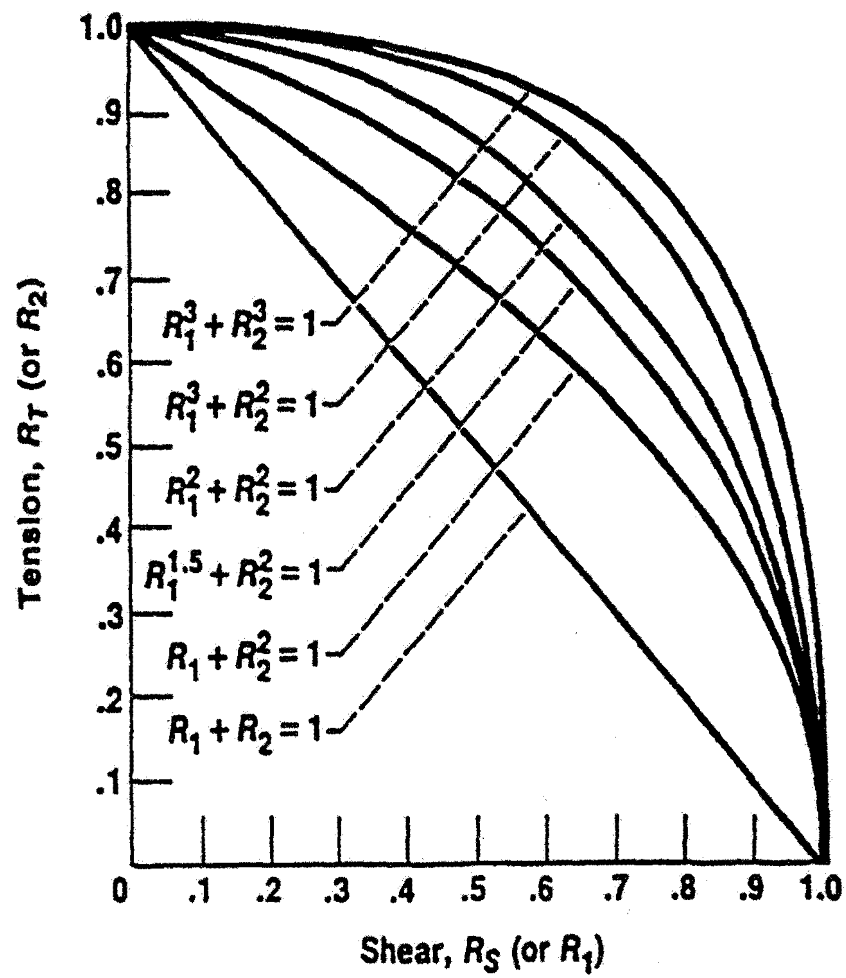
where X and y are exponents of the interaction curves shown in Figure 31. $R_S + R_T = 1$ is the most conservative and $R_S^3 + R_T^3$ is the least conservative.

Note that $R_S^X + R_T^y < 1$ is a requirement for a positive margin.

DESIGN CRITERIA

- **Combined Tension and Shear Loading** (Cont'd)

Figure 31 - Interaction Curves



DESIGN CRITERIA

- ## Fastener Spacing to Carry Horizontal Shear Loads

If we revisit the strength of materials section that deals with horizontal shear stresses in a beam $\left[f_s = \frac{VQ}{Ib}\right]$, we can use a systematic method to determine both the fastener size and spacing for a composite beam. This method is easier to show by example, rather than verbiage.

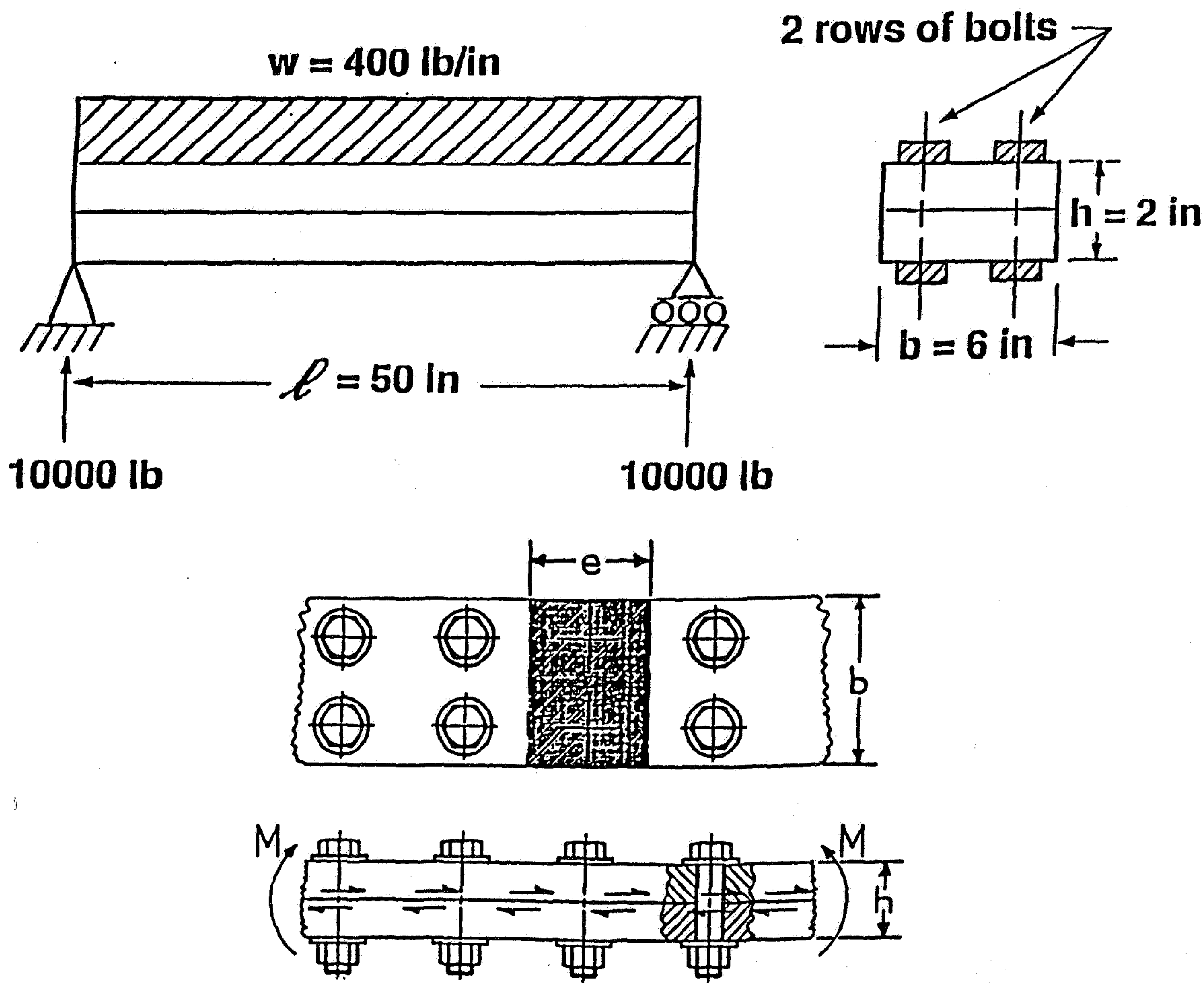
Using the beam diagrams of Figure 32 and reiterating:

h = 2 in. (2-1in. plates)
b = 6 in. width
w = load = 400 lbs./in. of length
ℓ = length of beam = 50 in. (simply supported)
V = vertical shear
Q = statical moment
I = Moment of inertia for beam cross section

DESIGN CRITERIA

Figure 32

Beam Loading for Horizontal Shear Load
(For this example: total beam length = 50 in.)



DESIGN CRITERIA

- **Fastener Spacing to Carry Horizontal Shear Loads (Cont'd)**

Reactions: $$R_1 = R_2 = \frac{400x50}{2} = 10000 \; lbs.$$

$$M = \frac{w\ell^2}{8} = \frac{400(50)^2}{8} = 125000 \; in-lbs.$$

e = bolt row spacing

Assume further that the plates are structural steel, with F_{tu} = 67 KSI and F_{ty} = 46 KSI (to carry beam bending stress)

A good guess for bolt diameter is 0.5 in., with clearance holes of 0.56 in. diameter.

Now $f_s = \dfrac{VQ}{Ib}$ *can be calculated.*

$$Q = \frac{bh}{2}\left(\frac{h}{4}\right) = [6x1](.5) = 3.0 \; in.^3$$

$$I = \frac{bh^3}{12} = \frac{6(2)^3}{12} = 4.0 \; in.^4$$

DESIGN CRITERIA

- **Fastener Spacing to Carry Horizontal Shear Loads (Cont'd)**

  Note:

  The moment of inertia (I) should be recalculated after the final bolt diameter is determined, deducting the area of the two bolt holes.

  Use: $I = \dfrac{(b-2D)h^3}{12}$ for the reduced area calculation.

DESIGN CRITERIA

- **Fastener Spacing to Carry Horizontal Shear Loads (Cont'd)**

For no bolt hole reduction:

$$fs = \frac{VQ}{Ib} = \frac{10000x3.0}{4.0x6.0} = 1250 \ psi$$

This stress will be across the entire shaded area of Figure 32, which is 6e. Then S(lbs.) = 6e(1250) = 7500e.

If we use two 0.5 dia. Grade 5 bolts (good for approx 10,500 lbs. each in shear without yielding),

$$e = \frac{2(10500)}{7500} = 2.8 \ in. \text{ max spacing between rows of bolts.}$$

DESIGN CRITERIA

- **Fastener Spacing to Carry Horizontal Shear Loads (Cont'd)**

This is too close, so we'll use 0.56 dia. Bolts with clearance holes of 0.63 dia.

Now $$I = \frac{(b-2D)h^3}{12} = \frac{[6-2(.63)]2^3}{12} = 3.17\ in^4$$

$$f_S = \frac{VQ}{Ib} = \frac{10000(3.0)}{3.17(6)} = 1577\ psi$$

$$S = f_S\,A_S = 1577(6e) = 9464e$$

Then $$e = \frac{2(13900)}{9464} = 2.94\ in.\ for\ row\ spacing$$

DESIGN CRITERIA

- **Fastener Spacing to Carry Horizontal Shear Loads (Cont'd)**

  This spacing is still close, but we won't go further since our goal was to illustrate the method of analysis.

  Beam bending and bearing stress calculations must still be done. The bolts must be critical in bearing to assure full shear load distribution.

  Note also that thin structures must be checked for "inter-rivet" buckling, and that the bolt spacing can increase as the vertical shear decreases.

DESIGN CRITERIA

- **Bolted Flanges with O-rings**

O-ring compression in a flange (see Figure 33) is normally only a small portion of the total bolt load. The O-ring groove is sized to give a specific range of O-ring compression when the flanges are metal-to-metal. For most O-rings, this compression value is 10% to 30% of the unloaded cross-sectional diameter. The flange surfaces in contact with the O-ring must be smooth to assure sealing without damage to the O-ring surfaces.

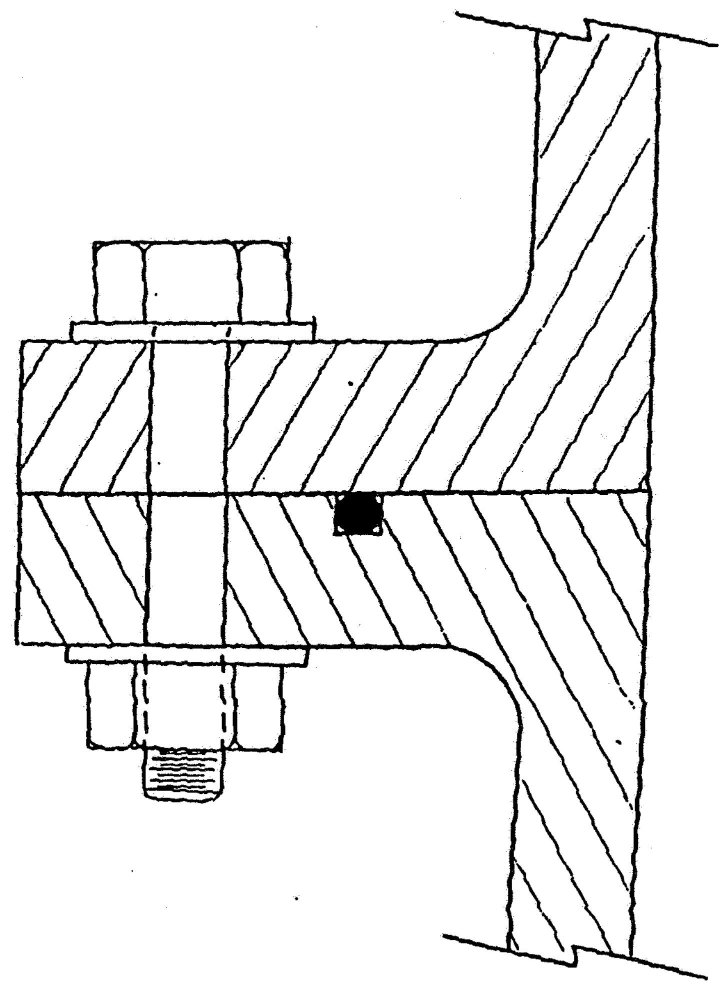
For O-ring installations, the fastener spacing must be close enough to keep the flanges from separating between fasteners.

A good design practice is to machine the O-ring groove in the cheaper of the mating flanges, as a groove machined too deep usually scraps the part. A dove-tail groove can be used if there is a concern for holding the O-ring in place during assembly/disassembly.

DESIGN CRITERIA

- **Bolted Flanges with O-rings** (Cont'd)

Figure 33 - Bolted O-Ring Joint



DESIGN CRITERIA

- **Bolted Flanges with Flat Gaskets**

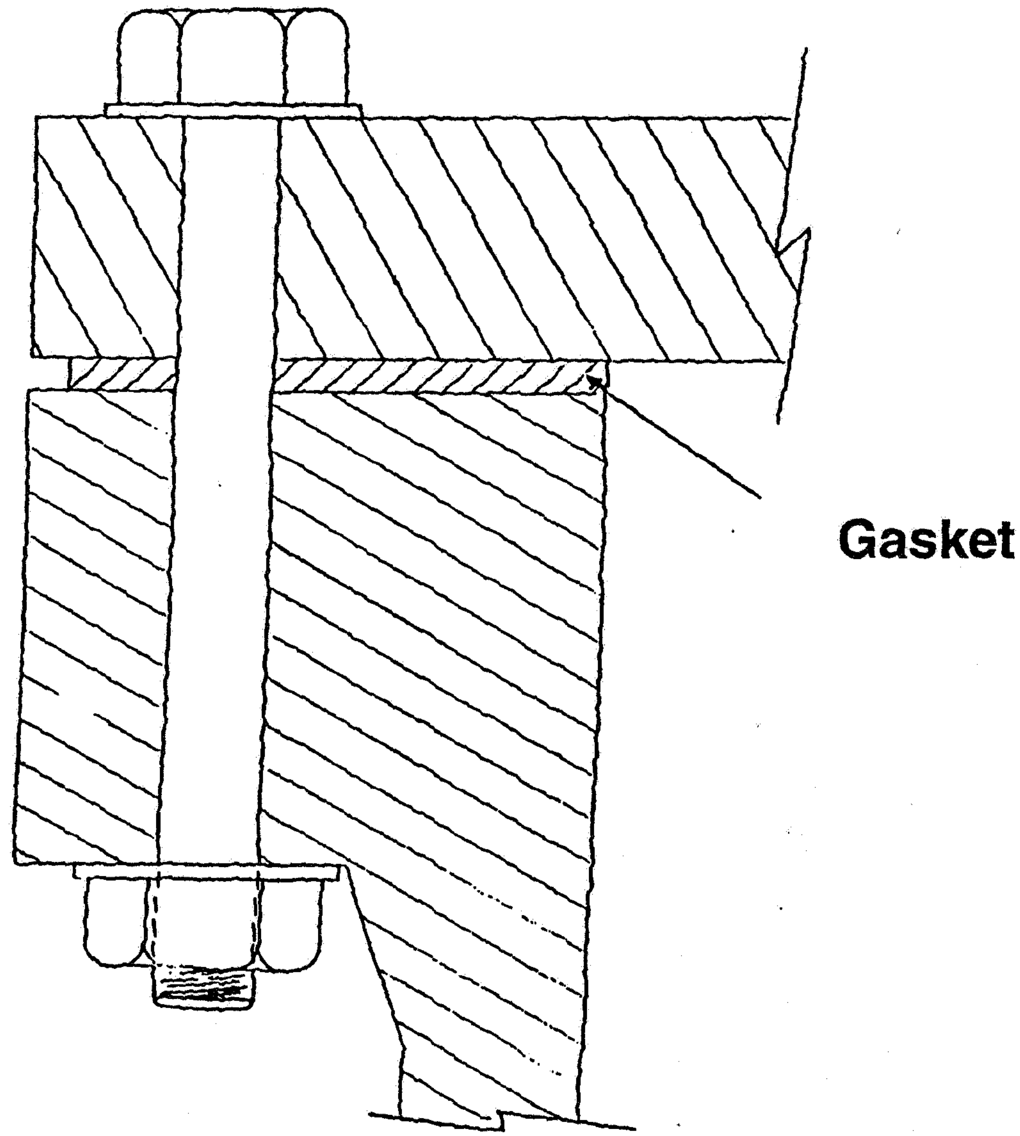
Bolt spacing and flange thickness are very critical for a flat gasket joint. If the flange can bow between bolts, the gasket can unload and leak.

Flat gaskets (See Figure 34) require minimal to heavy squeeze, depending on the pressure and the gasket hardness. Many gasket manufacturers give a compression value (psi or lbs. per linear inch) for their gaskets to seal. This sets a minimum bolt load, but care must be given to limiting the maximum bolt load to avoid crushing the gasket. Most gasket materials are not as heat resistant as the bolts, so safe room temperature bolt loads can yield a gasket at elevated temperatures.

Design practices to combat gasket leakage will be covered in subsequent pages.

DESIGN CRITERIA

Figure 34 - Bolted Flat Gasket Joint



DESIGN CRITERIA

- **Bolted Flanges with Flat Gaskets (Cont'd)**

  Some typical loading curves for flat gaskets are given in Appendix 4.

  Further information on flat gasket joint design can be found in J. Bickford's book (referenced earlier in this course).

  Gasket manufacturers usually have design data and design practices for the use of their gaskets.

DESIGN CRITERIA

- Gasket Loads in Flanged Joints

  - Leaks usually start at the points of maximum flange bending, which are usually midway between adjacent bolts where the gasket is not compressed enough to seal.

  - To increase the load at the critical midway point, three remedies should be considered:

    1. Increase the number of bolts.
    2. Increase the flange thickness.
    3. Increase the initial bolt torque.

DESIGN CRITERIA

- **Gasket Loads in Flanged Joints (Cont'd)**
  - Increased number of bolts
    - Flange deflection is proportional to the cube of the span between bolt centers.
    - Adding a bolt at midspan cuts the span between bolts by one-half and reduces the midspan deflection between bolts by a factor of eight, thereby increasing the gasket load.
    - Extra bolts improve the seal more than increasing bolt torque, but the cost is increased.

DESIGN CRITERIA

- **Gasket Loads In Flanged Joints (Cont'd)**
  - **Increased Flange Thickness**
    - Flange deflection is inversely proportional to the cube of the flange thickness.
    - Doubling the flange thickness decreases flange deflection by a factor of eight, thereby increasing gasket load at the critical midspan between bolts.
    - Increasing the flange thickness has a lesser cost effect than increasing the number of bolts, but the weight increases.

DESIGN CRITERIA

- Gasket Loads in Flanged Joints (Cont'd)
  - Increased Bolt Torque
    - Increasing bolt torque tends to increase flange bending; however, there is usually a slight increase in gasket loading at the point of maximum bowing which may be enough to close a leak.
    - If the bolt is already near its yield point, a further increase in torque cannot be made unless bolts with higher yield strengths are used or the bolt size is increased.

DESIGN CRITERIA

- **Bolted Flanges For Glass Windows**

Although glass windows are not usually associated with fastener design, there is the unique situation where a window must be installed in a pressure vessel. Then the fastener design becomes a balancing act to seal without overloading the window.

Glass is so brittle that it needs to be protected from direct contact with metal. This can be achieved with the sandwiching of the glass in flat rubber gaskets with a bumper strip around the outer periphery of the window to keep it from direct contact with metal. Since the glass has approximately one sixth the thermal expansion/contraction coefficient of most metals, the flat gasket must carry this dimensional variation and still seal without overloading anything.

DESIGN CRITERIA

- **Bolted Flanges for Glass Windows** (Cont'd)

  A typical pressure vessel window design is shown in Figure 35. Note that the machining tolerances for both the bottom and top gasket faces must be factoring into the total gasket compression range. Note also that the retaining ring is metal-to-metal, so the depth of the window and gasket well must be controlled to seal without exceeding the seal allowables.

DESIGN CRITERIA

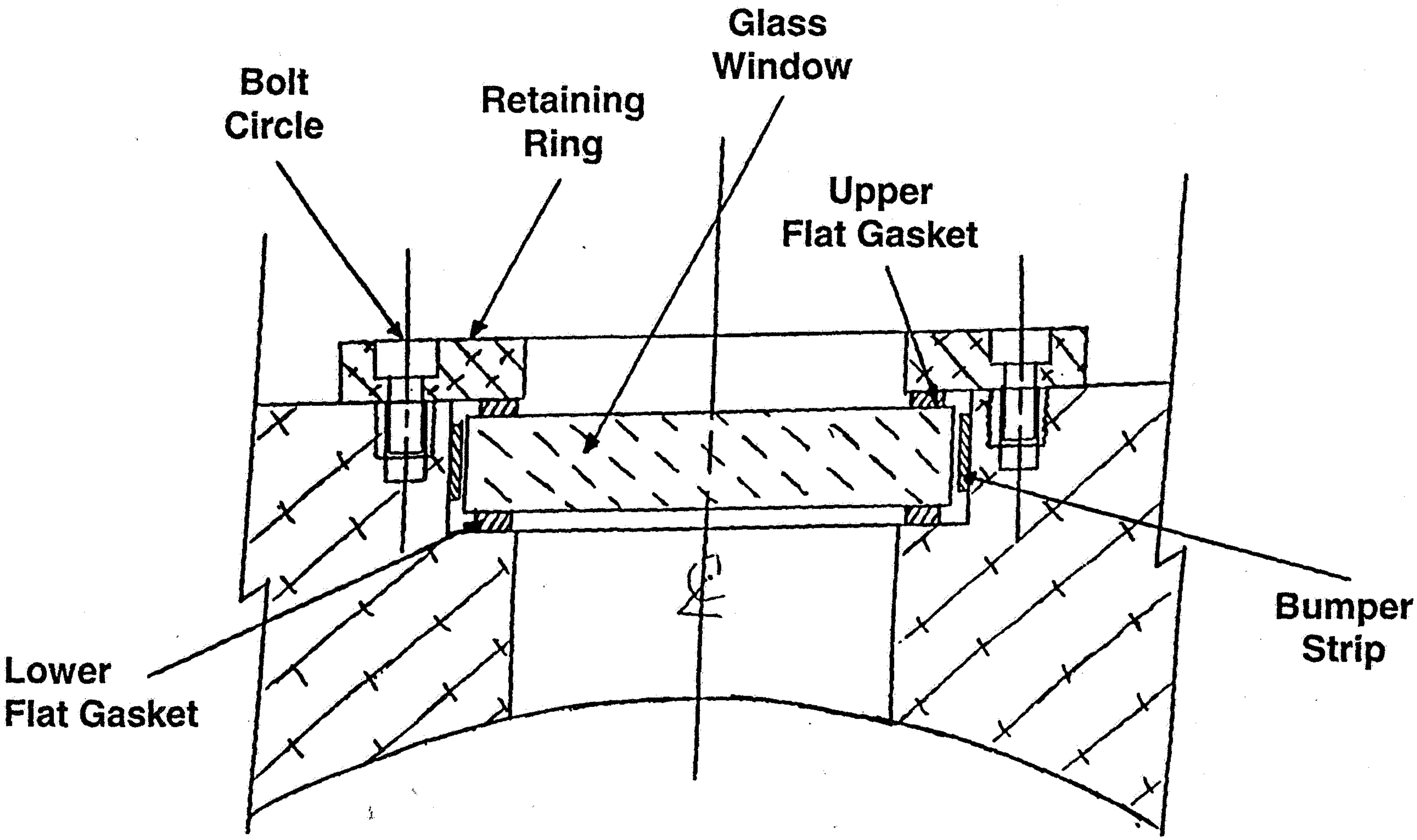


Figure 35 - Window Design For a Pressure Vessel

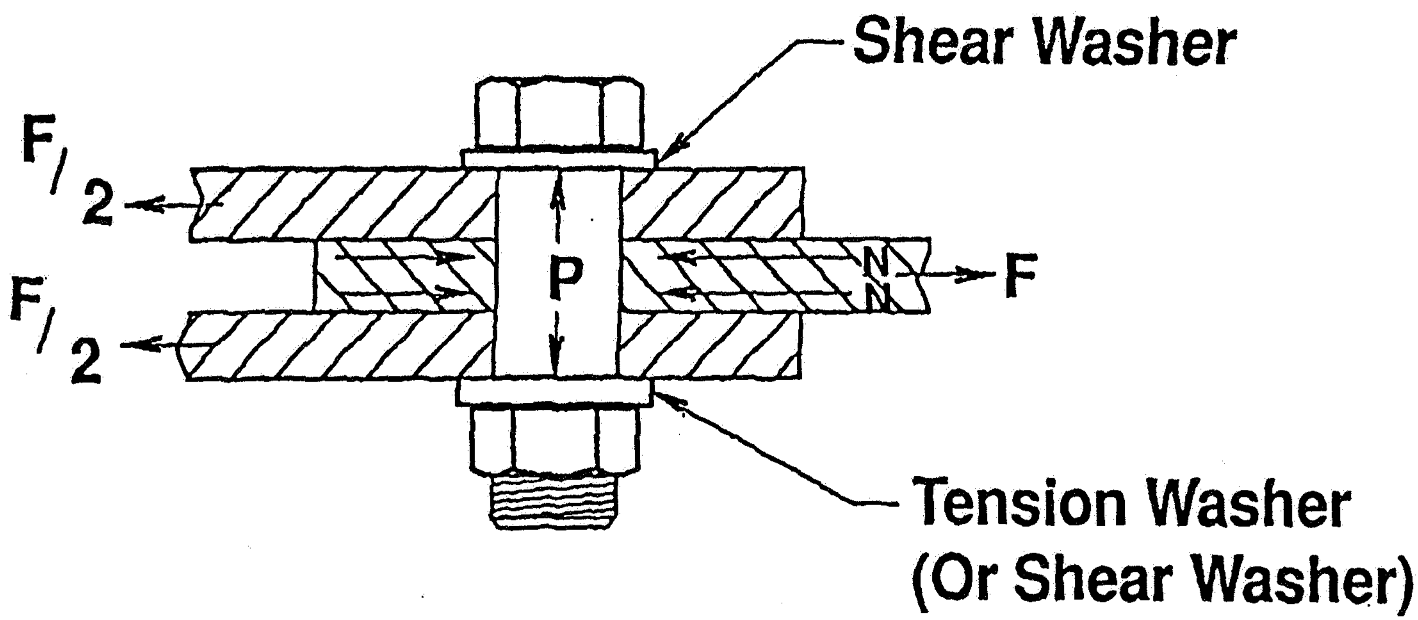
DESIGN CRITERIA

- **The Effect of Friction in a Clamped Joint**

In most cases, friction forces between clamped (bolted) surfaces are not included in shear load capabilities. The reason for not including friction forces is the inability to determine the coefficient of friction between clamped surfaces and accurate clamping forces generated by each fastener. (See Figure 36 for forces due to joint friction).

In some buildings and bridges, this contact friction is monitored closely enough to be included in the joint reaction forces. However, the bolts are torqued to near yield values in most cases to maximize friction forces.

DESIGN CRITERIA



P = Bolt Preload

N = Friction Load = Pμ

Shear Load on Bolt = F-2N

Figure 36 - Friction Forces In A Bolted Joint

DESIGN CRITERIA

- **Compression "Cone" of a Bolted Joint**

  Some of this information and supporting analysis was given earlier in the joint stiffness section for calculating cone stiffness.

  Additional methods of calculating compression stiffnesses are given in Appendix 3.

  Comparisons of the relative conservatism for the spherical, cone, and cylindrical models are also given in Appendix 3.

  Note that in most cases the bolt/joint relative stiffness calculations are not required. Satisfactory stiffness ratios can be selected by design experience and common sense for most designs.

DESIGN CRITERIA

- **Bolting of Dissimilar Materials**

Dissimilar materials can be a source of many problems in a bolted joint. Some of these problems are:

1. Differential thermal expansion and contraction can cause bolt overload or loosening. Be sure to check the joint at both ends of its temperature range.

2. Galvanic corrosion is possible unless mating surfaces are insulated from each other.

3. Yielding of the softer materials must be checked by analysis.

4. Material strengths at temperature extremes must be checked to avoid yielding.

DESIGN CRITERIA

- **Maximizing Effective Length of Fasteners**

When we previously discussed stiffness ratios, the effective length of a fastener was mentioned. This is particularly important where differential expansion/contraction is present.

It may be necessary to add a spring or Belleville type washers under a bolt head to increase its effective length enough to satisfy the design. Remember that the deflection formula for the fastener is:

$\Delta L = \dfrac{PL}{AE}$, so increasing L gives more adjustment capability.

DESIGN CRITERIA

- ## Match Drilling of Fastener Holes

Match drilling of mating holes is always desirable, but it should be mandatory for fasteners in shear. Match drilling is accomplished by having a pilot hole in one part. The mating parts are clamped in position. Then the pilot hole is drilled through the entire thickness to the desired diameter. This gives precision matched parts, even if the holes are not in their true positions.

Some examples of mismatched countersunk holes in shear are shown in Figure 37.

Note that the mismatch usually produces fastener head bending.

DESIGN CRITERIA

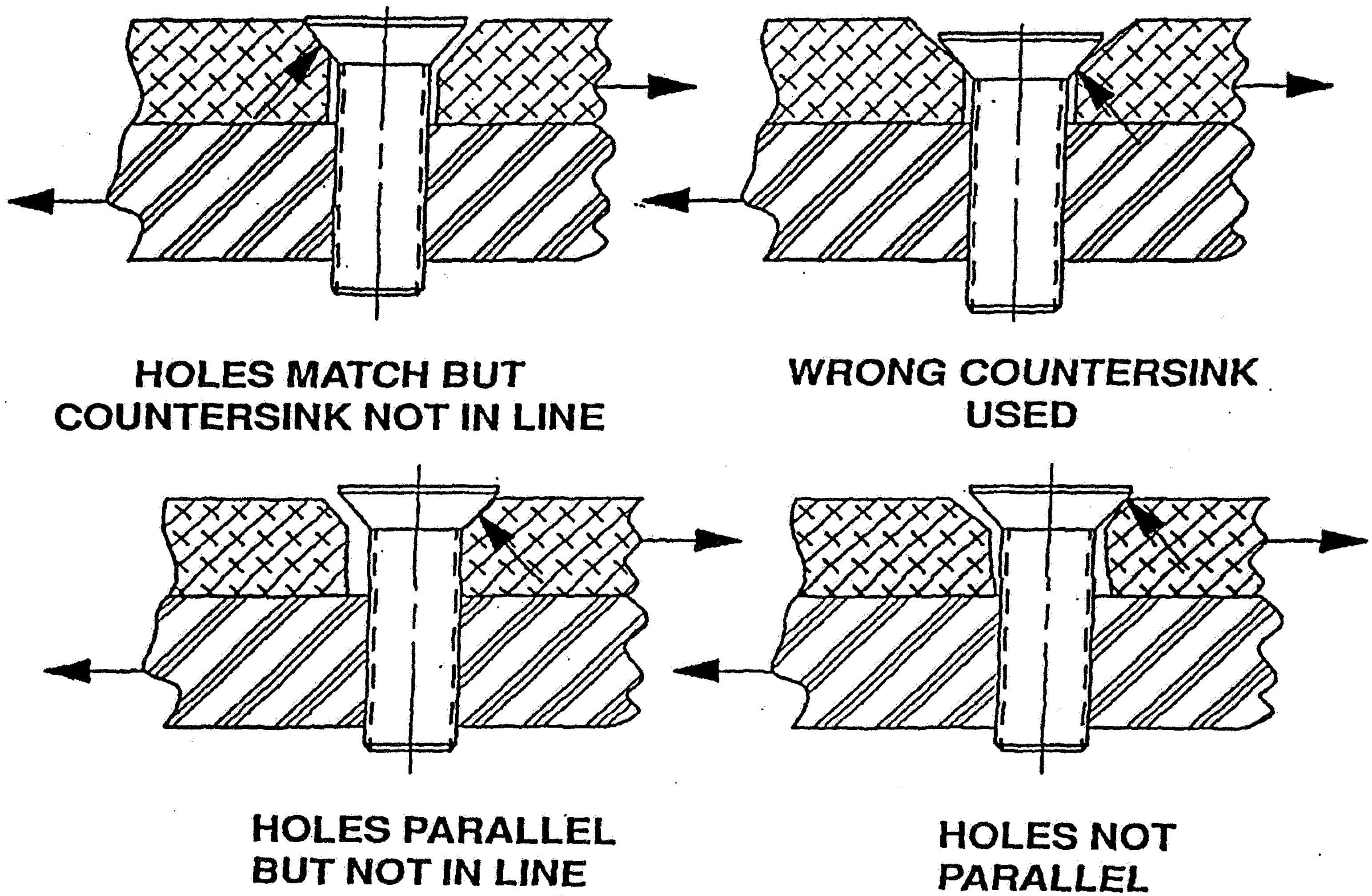


Figure 37 - Shear Loading of Mismatching Countersunk Fasteners

DESIGN CRITERIA

- ## Knife Edges in a Countersunk Hole

Knife edges are stress risers and are to be avoided in countersunk sheet holes (see Figure 38 A). To avoid this problem, make “t” greater than 1.5h as shown in Figure 38 B.

If it is not possible to have ‘t’ greater than 1.5h, dimple the outside sheet and countersink the hole in the inner sheet as shown in Figure 38 D.

If neither sheet is thick enough to machine a countersunk hole, both sheets can be dimpled to provide a flush outside surface as shown in Figure 38 C. However, the sheets must have good ductility to dimple without cracking.

Note that most aerospace companies do not allow dimpling on primary structure, due to its unpredictable fatigue life.

DESIGN CRITERIA

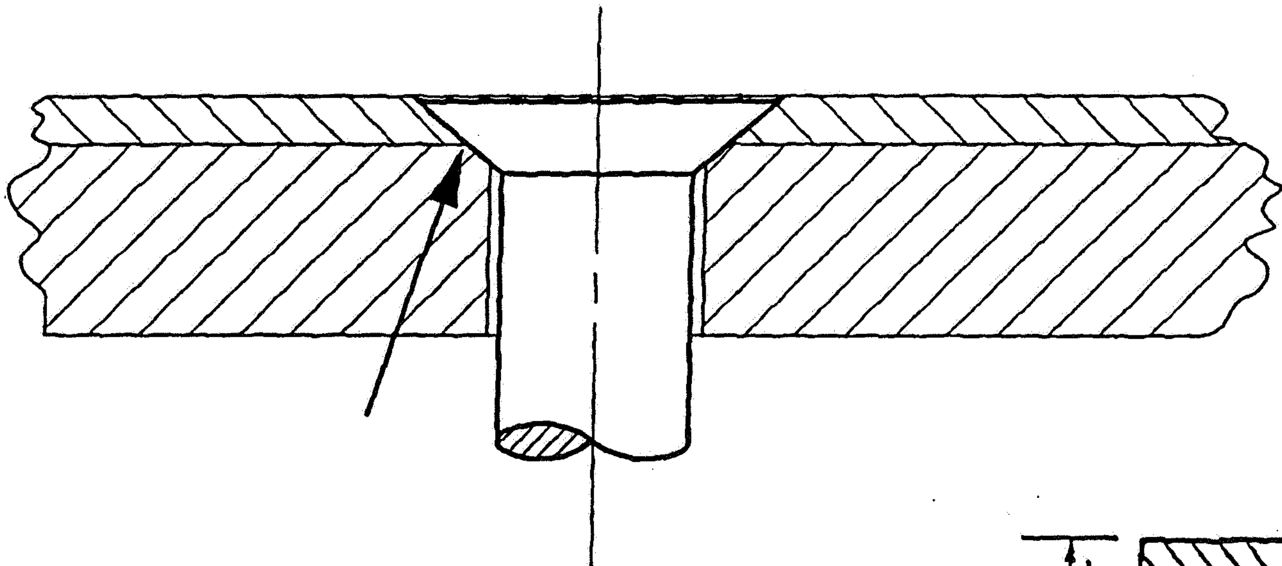


Figure A

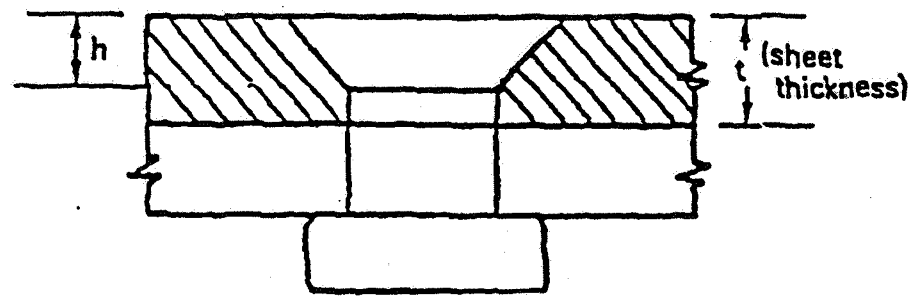


Figure B

Figure 38 - Knife Edge in Countersunk Sheet

DESIGN CRITERIA

Both Holes Dimpled

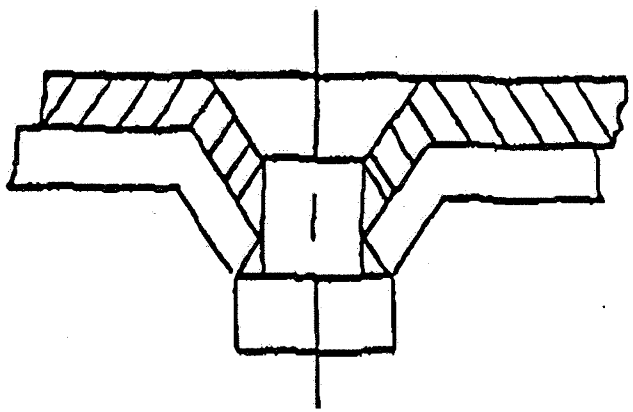


Figure C

Dimpled and Countersunk Holes

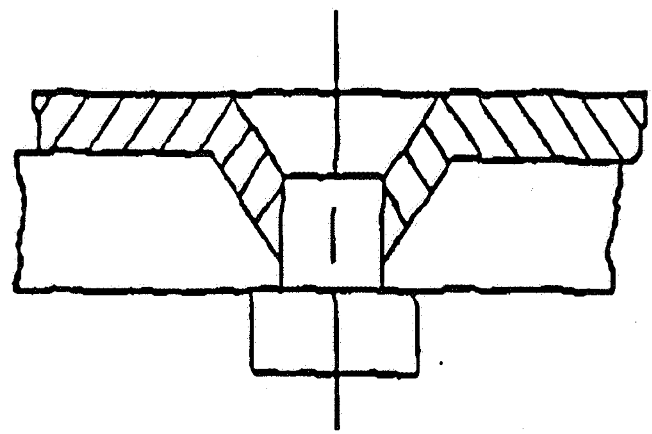


Figure D

Figure 38 - Knife-Edge in Countersunk Sheet

DOWEL PINS

- ## Introduction

Dowel pins are close tolerance pins which are used to align two mating components. The pins are usually mounted in one piece with a slight interference fit. The mating piece has close tolerance holes to slip over the protruding pins. Then the pieces are bolted together and the bolts are analyzed for the total shear load. The dowel pins should NOT be included with the bolts in shear load calculations.

DOWEL PINS

- **General Design Criteria**

Dowel pins can be designed to carry all of the shear loads. The difference in fit tolerances between bolts and dowel pins will allow the pins to load up before the bolts can pick up any shear loads, so the load carrying is either pins or bolts.

Dowel pins in blind holes should be avoided if pin removal is required. It is much better to have a through hole which allows the pin to be driven out from the back side. A vented pin (with groove or flat edge) is desirable for blind installations.

DOWEL PINS

- **General Design Criteria** (Cont'd)

Tapered dowel pins are available, and pins with external serrations or ridges (to prevent pin rotation after installation) are also available.

Shear allowables for dowel pins are usually determined by manufacturer's test programs, since the area at the shear plane may vary. (The pin cross section may not be constant and/or a common geometric shape).

Some of the common types of dowel pins are shown in Figures 39 through 43.

DOWEL PINS

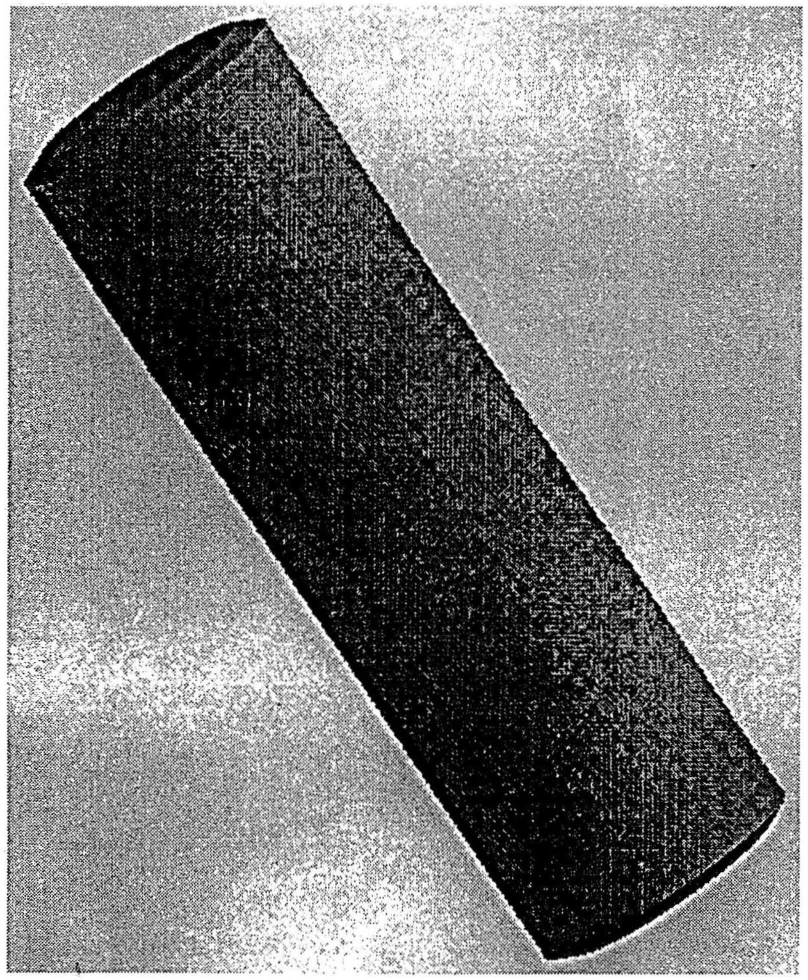


Figure 39
Straight (SOLID) Dowel Pin

DOWEL PINS

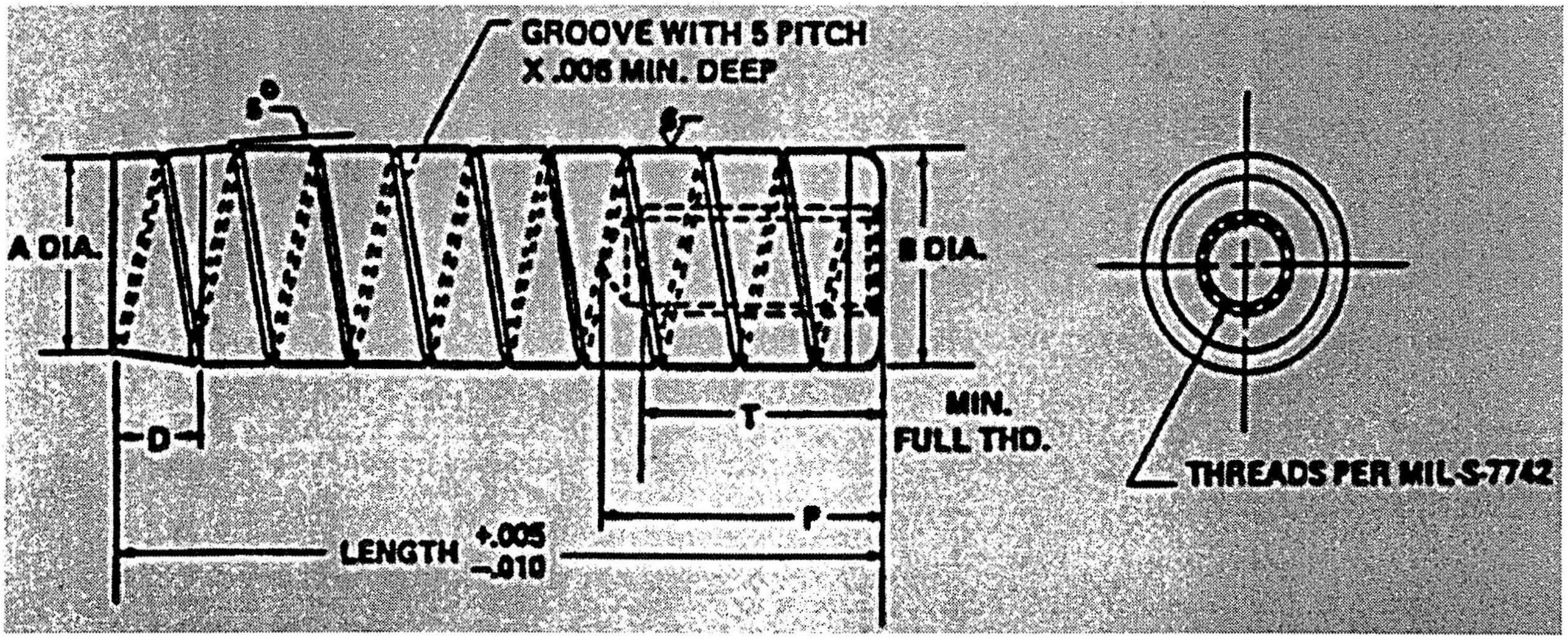


Figure 40
Drilled And Tapped Dowel Pin with Vent

DOWEL PINS

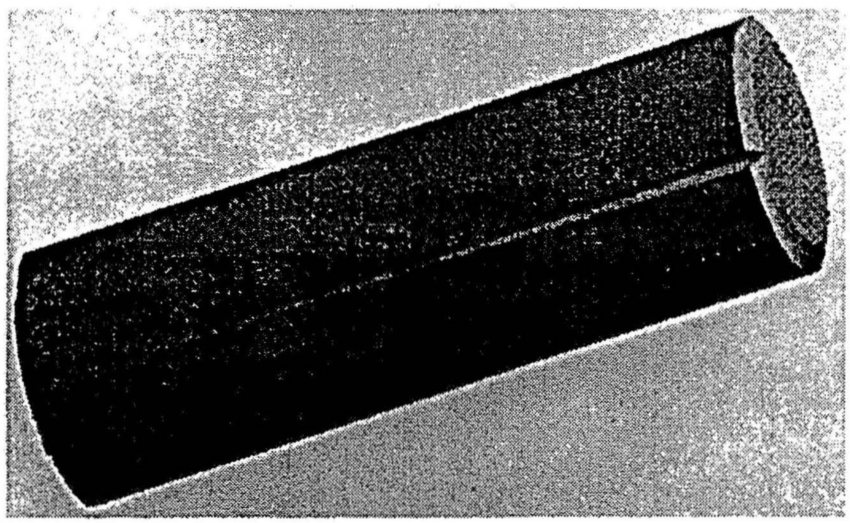


Figure 41
Grooved Dowel Pin

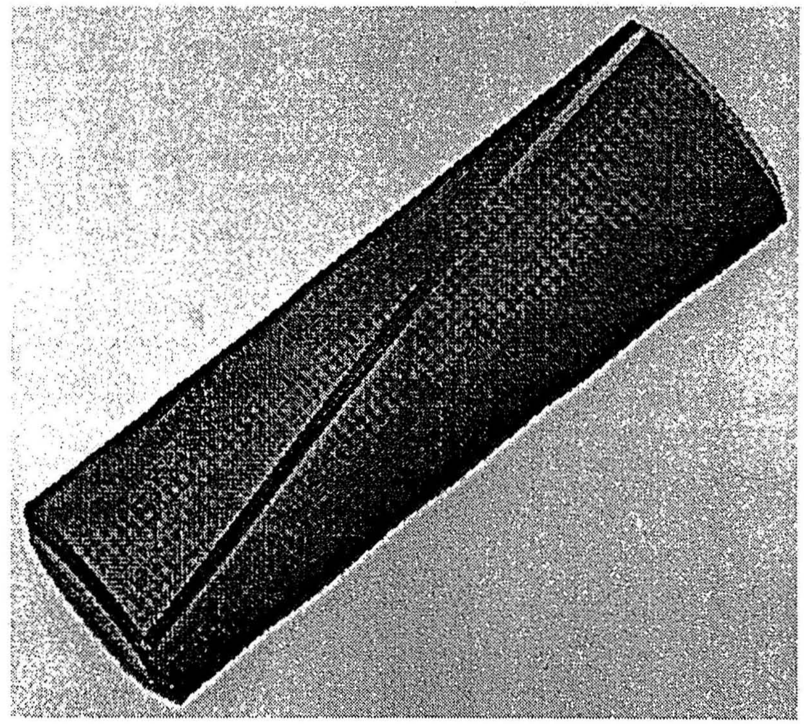


Figure 42
Vented Dowel Pin

DOWEL PINS

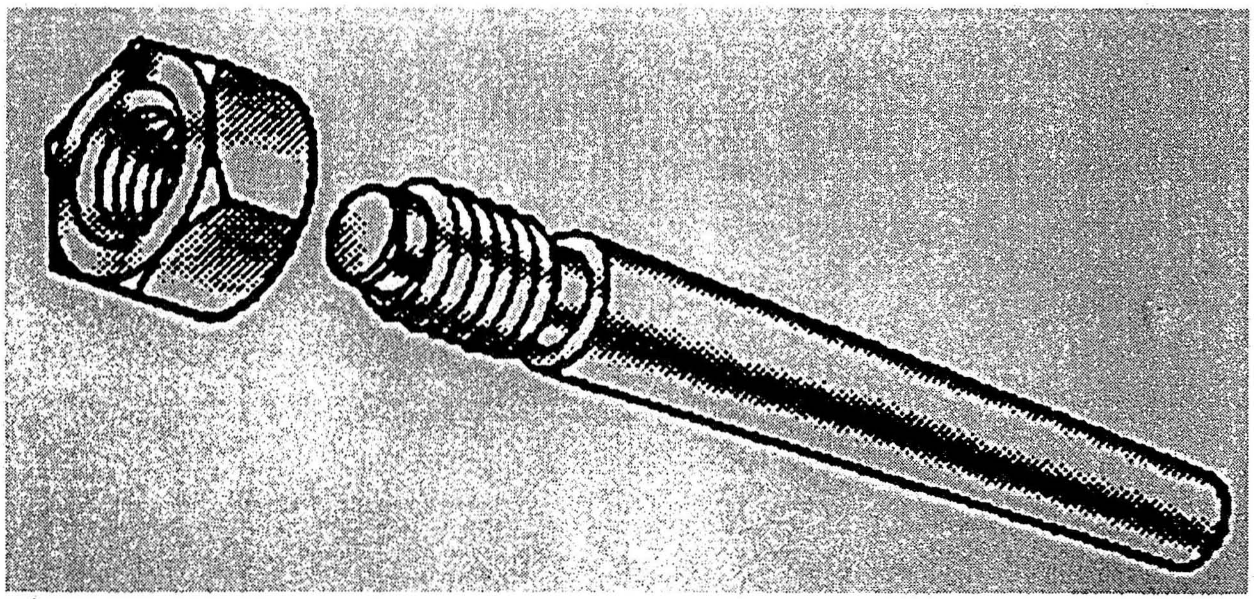


Figure 43
Tapered Dowel Pin With Jacking Nut

ROLL PINS

- **General Type**

Roll pins (also called spring pins) are usually made by rolling a piece of thin alloy steel sheet to a given diameter, with a chamfer on each end. It is then heat treated to a high hardness. The chamfer on the ends allows it to be started and driven into a hole. The coiled cross section decreases in diameter during driving to give an interference fit. See Figure 44 for a typical roll pin.

- **Slotted Tubular Pin**

If a sheet is rolled into a slotted cylindrical cross section, with no overlap, it is called a slotted tubular pin. See Figure 45 for a slotted tubular pin.

ROLL PINS

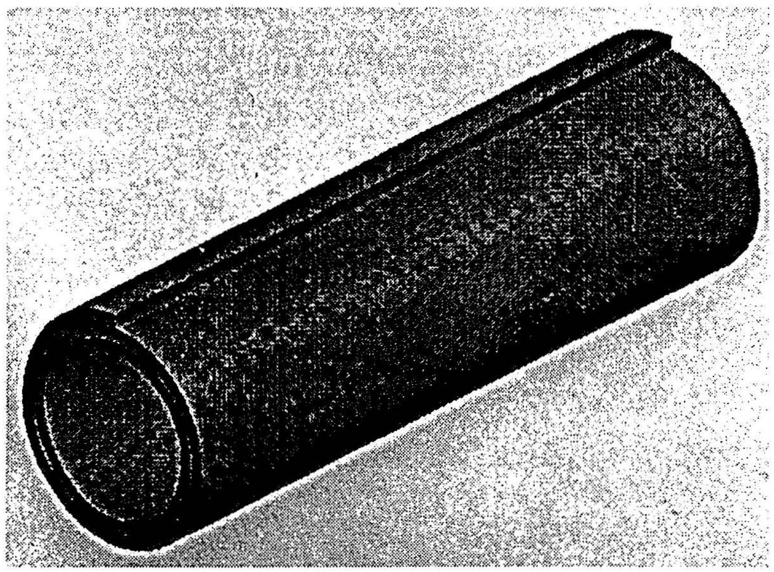


Figure 44

Roll Pin

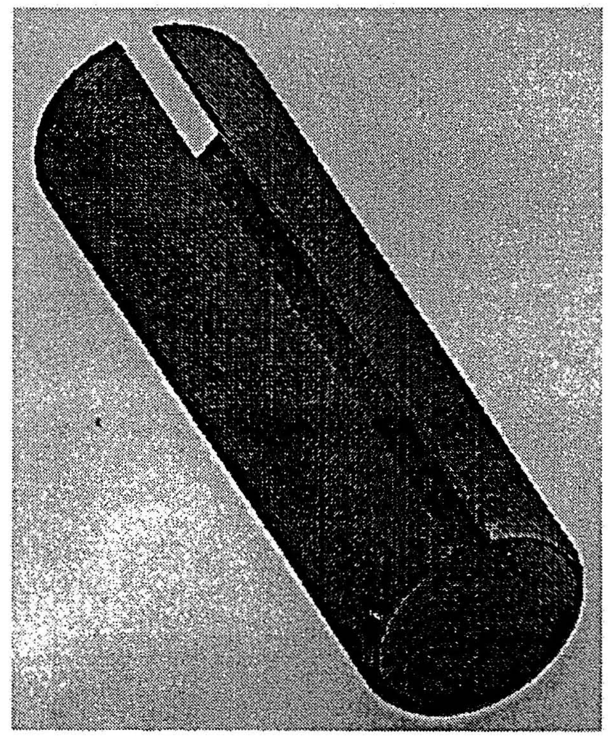


Figure 45

Slotted Tubular Pin

ROLL PINS

- **General Usage**

Roll pins and slotted tubular pins are used for quick-disconnect interference joints, such as attaching a handle to an actuating shaft. They can also be used as a fixed pin in a rotating joint where the rotating component has an oversize hole to allow free rotation around the pin.

Load carrying capabilities for both roll pins and slotted tubular pins are usually determined and tabulated by the pin manufacturer.

RIVETS

- ## Introduction

Rivets are relatively low cost, permanently installed fasteners that are lighter weight than bolts. As a result, they are the most widely used fasteners in the aircraft manufacturing industry.

Rivet installation is usually faster than bolt installation, since automatic installation tools can be used for riveting.

Rivets work best in thin sheet designs where shear is the dominant load, since rivets don't have good tensile capacity. Rivets should be designed to be bearing critical, as they usually are loaded as a group. The longer the grip length (total thickness of sheets being joined) the more difficult it becomes to lock the rivets.

RIVETS

- **Introduction** (Cont'd)

Although rivets are installed with an interference fit, they are neither airtight nor watertight. Sealants must be applied to the joints if sealing is required.

Since rivets are permanently installed, they must be removed by drilling or punching them out and replacing them with oversized rivets. This is a laborious task.

RIVETS

- ## Rivet Materials

Rivets are made of various carbon steels, CRES, brass, aluminum, monel, and titanium. These materials must be ductile enough to deform to their final configuration without cracking. Since fairly high strength is required of these rivets , the strength vs. ductility is a metallurgical balancing act. A list of common aerospace rivet materials is given in Table 13.

Many rivets contain more than one material. This enables tailoring the rivet to the joint material it is in.

RIVETS

Table 13 - Aluminum and Other Solid Rivet Materials

Material	Rivet Designation	Rivet Heads Available	Applications
2117-T4	AD	Universal (MS20470) 100° Flush (MS20426)	General use for most applications
2024	DD	Universal (MS20470) 100° Flush (MS20426)	Use only as an alternative to 7050-T73 where higher strength is required
1100	A	Universal (MS20470) 100° Flush (MS20426)	Nonstructural
5056-H32	B	Universal (MS20470) 100° Flush (MS20426)	Joints containing magnesium <u>only</u>
Monel (Annealed)	M	Universal (MS20615)	Joining stainless steels, titanium, and inconel
Copper (annealed)	---	100° Flush (MS20427)	Nonstructural
7050-T73	E	Universal (MS20470) 100° Flush (MS20426)	Use only where higher strength is required

RIVETS

- ## Head Types

There are many head types for rivets. Some of the most common types, particularly for solid rivets, are shown in Figure 46. In spite of the desire to standardize, each rivet manufacturer has his own rivet geometry.

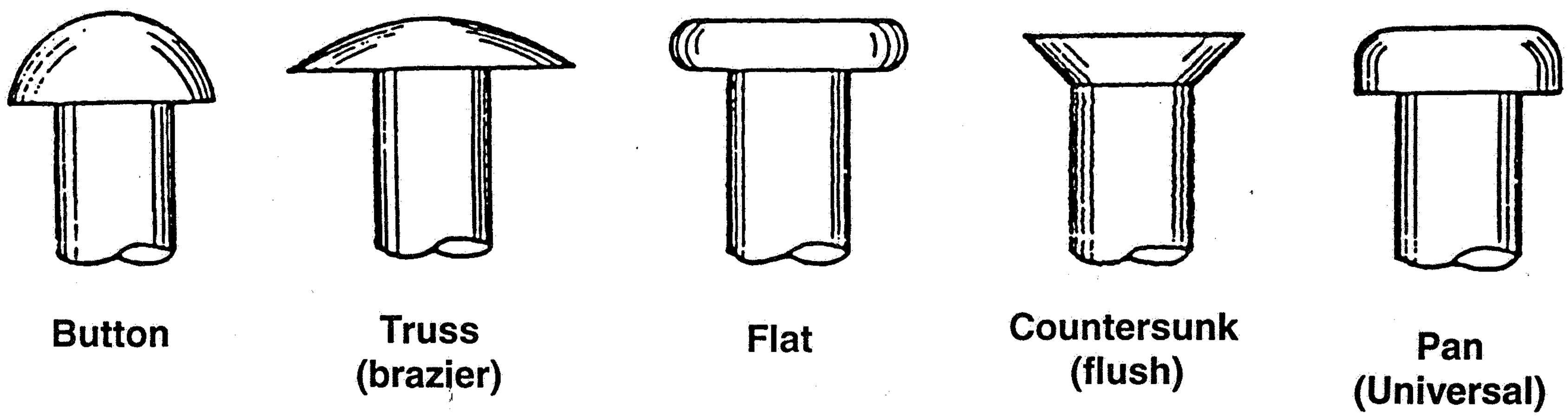


Figure 46 - United States Standard Rivet Heads

RIVETS

- **Solid Rivets**

Solid rivets are made for both construction and aerospace applications. Since their methods of installation are different, I'll cover them separately.

Solid rivets for construction are large diameters (.313 through 2.00 in.), and made of steel. They can't be installed cold, so they are pre-heated to approximately 1800°F at installation. In the past, most bridge trusses and portal bracing were riveted. Now very little riveting is done in the construction field. Welding or bolting is cheaper and faster in most cases. (Construction rivets are covered in ASTM A502).

RIVETS

- **Solid Rivets** (Cont'd)

Solid rivets for aerospace usage are small diameters (.125 through .250 in.) and are installed at room temperature. One type (2024-T4 aluminum) must be kept at 0°F until installation to prevent cracking after installation.

Solid rivet installation requires both ends of the rivet to be accessible for a pneumatic hammer and a bucking bar.

RIVETS

- ## Blind Rivets

Blind rivets get their name from the fact that they can be completely installed from one side. They have the following advantages over solid rivets:

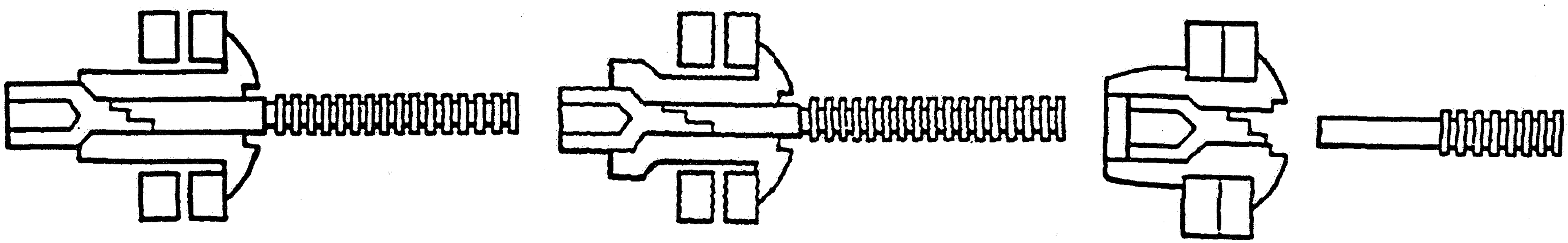
1. Only one operator is required for installation.
2. The installation tool is portable (comparable to an electric drill in size).
3. Only one side of the workpiece needs to be accessible.
4. A given-length rivet can be used for a range of material thicknesses.
5. Installation time is faster than for solid rivets.
6. Clamping force is more uniform, since it is controlled by a machine.
7. Less operator training is required.

RIVETS

- ## Specific Blind Rivets

  ➣ **Pull Mandrel**

  This rivet is installed with a tool that applies force to the rivet head while pulling a pre-notched serrated mandrel through to expand the far side of the tubular rivet to form a head. When the proper load is reached, the notched mandrel breaks as shown in Figure 47.



Rivet Inserted | Start Setting | Assembly Complete

Figure 47 - Pull Mandrel Rivet Installation

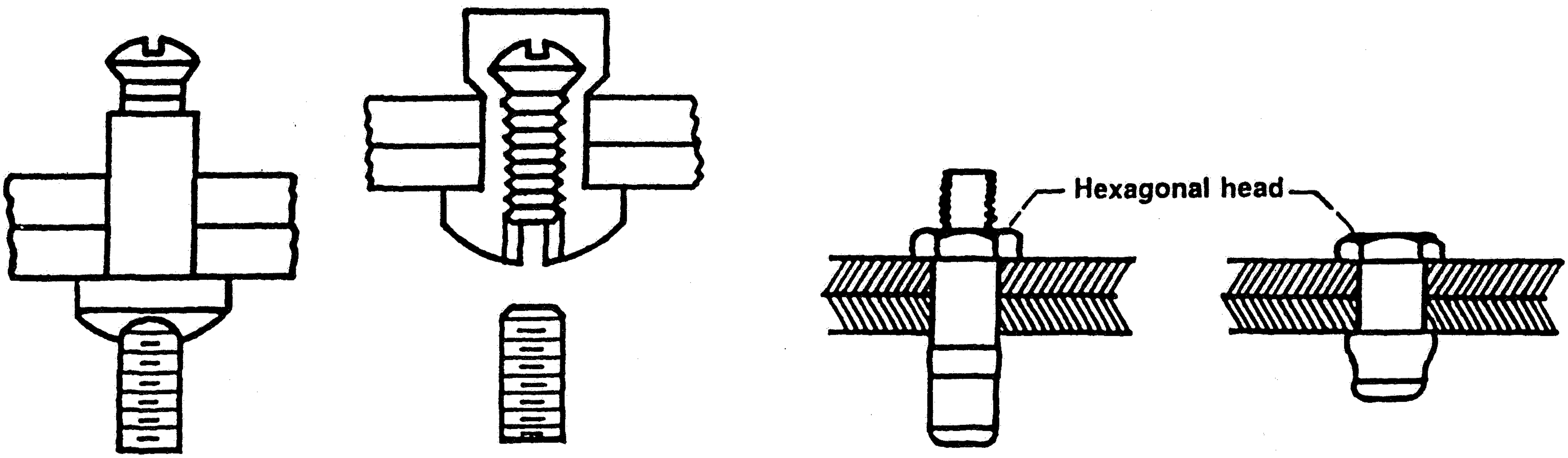
RIVETS

- ## Specific Blind Rivets (Cont'd)

  ➛ **Threaded Stem Rivet**

  This rivet has a threaded mandrel (stem) with the external portion machined flat on two sides for the tool to grip and rotate. The head of the tubular body is normally hexagonal to prevent rotation while the mandrel is being torqued and broken off. Two types of this rivet are shown in Figure 48.

RIVETS



Inserted Installed Inserted Installed

(a) (b)

(a) One-piece body
(b) Two-piece body

Figure 48 - Threaded-Stem Rivets

RIVETS

- ## Specific Blind Rivets (Cont'd)

  ➛ **Drive-pin Rivet**

  This rivet has a pin that is driven through the far side of the rivet (which is cut in segments) to form a shop head (as shown in Figure 49). Although these rivets are easy to install they are used primarily in industrial sheet metal joining.

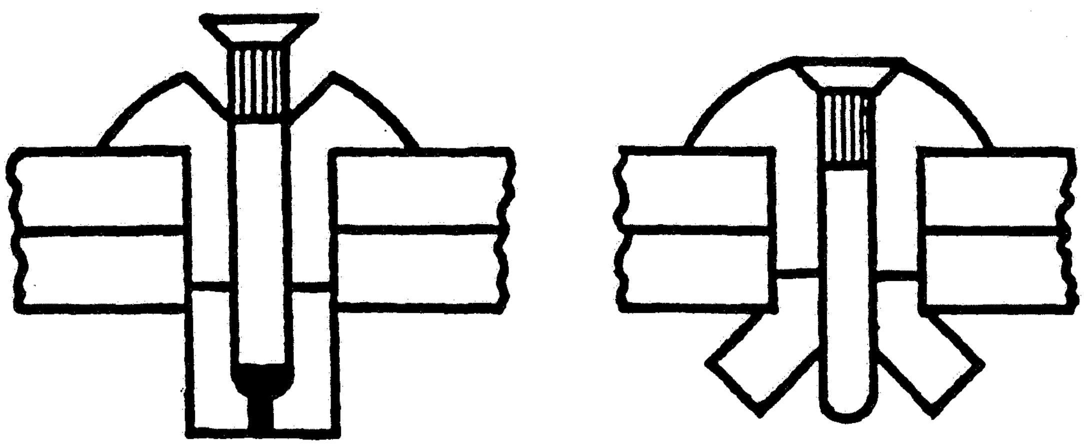


Figure 49 - Drive-pin Rivet

RIVETS

- ## Full Tubular

A full tubular rivet is very similar to the semi-tubular rivet, except that the hole on the field end is deeper. It is a weaker rivet than the semi-tubular rivet, but it can be driven through soft materials prior to clinching (see Figure 50).

This rivet is used primarily for non-critical industrial applications.

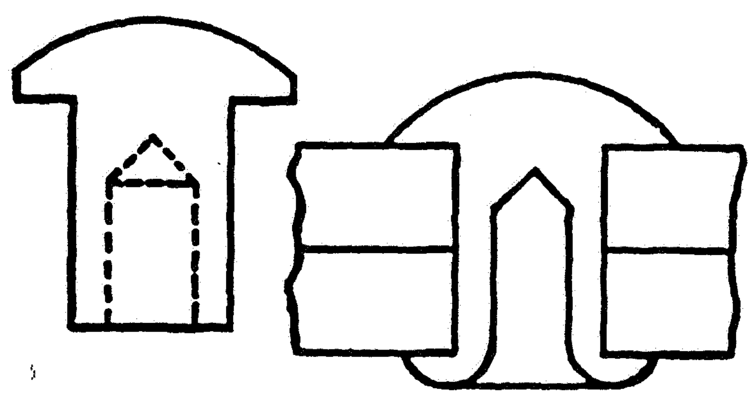


Figure 50 - Full Tubular Rivet

RIVETS

- ## Semitubular

A semitubular rivet is solid except for a hole (hole depth up to 1.12 of shank diameter) in the field (shop) end. When the rivet is installed, it approaches solid rivet configuration (as shown in Figure 51).

Note that this type of rivet must be ductile to form without cracking. Therefore it can't be made from a high strength material.

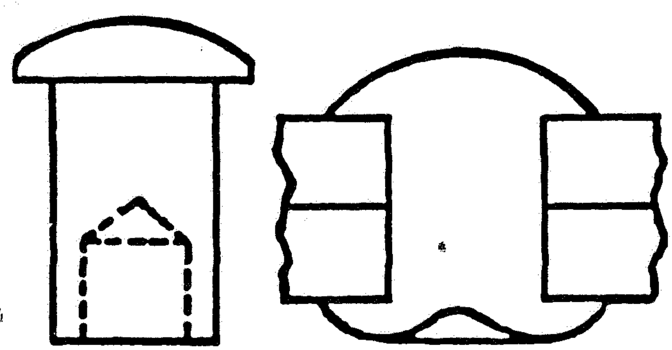


Figure 51 - Semitubular Rivet

RIVETS

- ## Metal Piercing

Metal piercing rivets are similar to semitubular rivets, except that they have greater column strength. The hole in the field side is not drilled all the way through. The rivet mushrooms out as it pierces and locks in place (see Figure 52).

Once again, this type of rivet is used for industrial applications, rather than for aerospace.

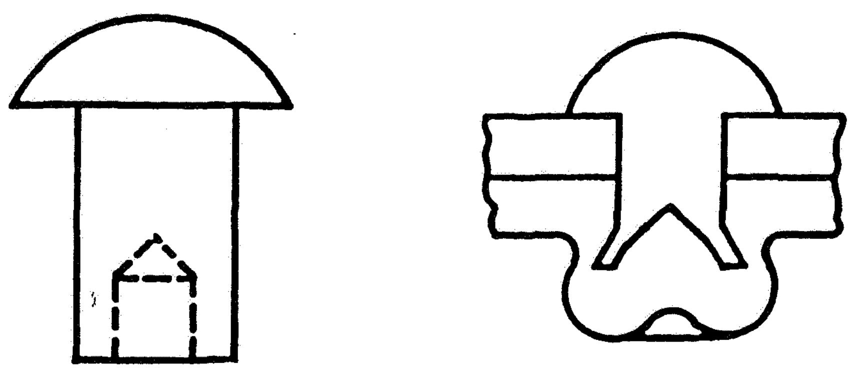


Figure 52 - Metal Piercing Rivet

RIVETS

- ## Split (bifurcated) Rivet

This is the old-fashioned copper rivet used by farmers to mend harness. The rivet is hard enough to drive through a material such as leather but is soft enough to clinch without breaking. There is a wire holder available to hold the rivet in place while it is being pounded (see Figure 53).

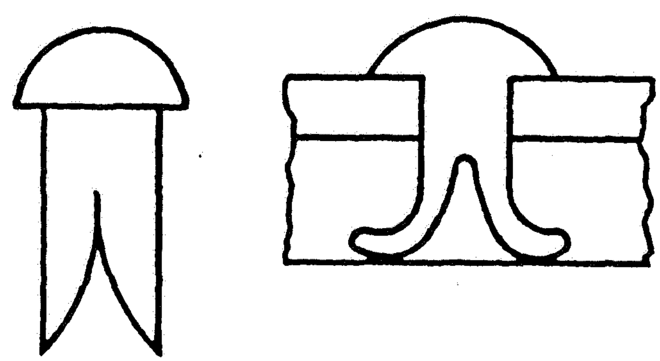


Figure 53 - Split (bifurcated) Rivet

RIVETS

- ## POP Rivets

POP rivets (now Black & Decker) are blind rivets used primarily for home repairs. They have a “nail” type stem which is gripped by a hand held gun and pulled through the back side of the tubular body to expand it into a shop head (see Figure 54).

The stem sometimes falls out of the body after installation, and the symmetry of the shop head is not too good. This rivet is not recommended for critical structures.

RIVETS

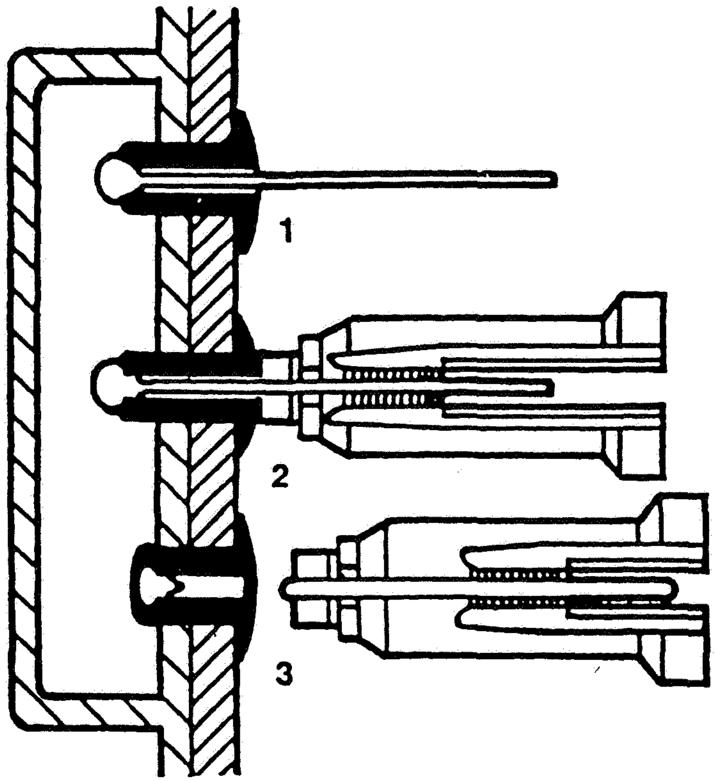


Figure 54 - POP Rivet Installation

Rivets (cont'd)

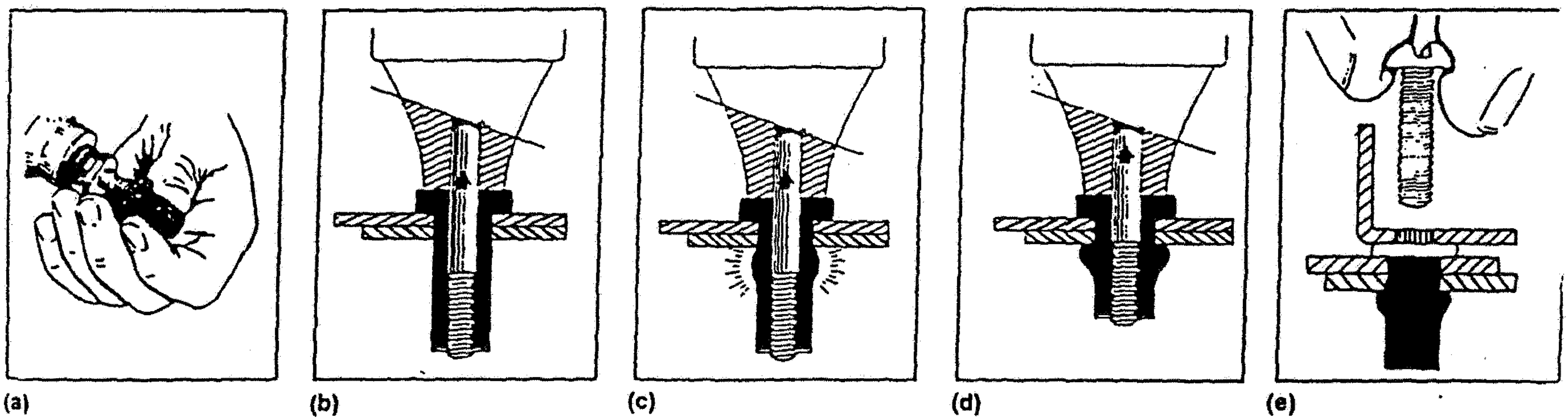
- ## Rivnuts

A Rivnut (BF Goodrich) is a tubular rivet with internal threads. It is deformed in place in a hole and becomes a blind nutplate. (See Figure 55 for the installation procedure).

Since the unthreaded tubular portion of the rivet must deform, the material must be ductile. Consequently, Rivnuts do not have high strengths.

Rivnuts are available with protruding, countersunk, and fillister heads. A variety of ends and shanks are also available.

RIVETS



(a) Step 1--Rivnut fastener is threaded onto mandrel of installation tool.
(b) Step 2--Rivnut fastener, on tool mandrel, is inserted into hole drilled for installation.
(c) Step 3--Mandrel retracts and pulls threaded portion of Rivnut fastener shank toward blind side of work, forming bulge in unthreaded shank area.
(d) Step 4--Rivnut fastener is clinched securely in place; mandrel is unthreaded, leaving internal Rivnut threads intact.
(e) Blind nutplate--Properly installed Rivnut fastener makes excellent blind nutplate for simple screw attachments; countersunk Rivnut fasteners can be used for smooth surface installation.

Figure 55 - Rivnut Installation

RIVETS

- **AD and DD Solid Rivets**

  ➢ The most common aerospace rivets are the AD and DD solid aluminum rivets, as listed in Table 14. These are the preferred rivets for joining aluminum structures. Keep in mind that the design range for an airplane skin can vary between -65°F and 140°F, so it is desirable to have rivets and structure with the same thermal coefficients.

  ➢ The "icebox" (DD) rivets can be used in higher strength applications, but they must be kept at 0°F until they are installed (to form heads without cracking). The 7050-T73 aluminum rivets are an alternative to "icebox" rivets.

RIVETS

- **AD and DD Solid Rivets (Cont'd)**

  - ➛ **5056 aluminum rivets are stress corrosion sensitive in all materials except magnesium.**

  - ➛ **Since solid rivets are expanded to an interference fit, they *SHOULD NOT* be used in composite materials. The hoop tension in a composite material hole can cause delamination of the material surfaces.**

RIVETS

- ## Monel Rivets

  - ➢ Monel is an alloy which contains 67% nickel and 30% copper. It is stronger (F_{SU} = 49ksi) and more heat resistant than aluminum. Yet it is ductile enough to cold form as a solid rivet without cracking.

  - ➢ Monel rivets are used for joining stainless steels, titanium, and Inconels. Monel should not be used for joining aluminum.

- ## Titanium/Columbium Rivets

  - ➢ These (45Cb alloy) rivets (per MIL-R-5674 and AMS4982) have a shear strength of 50ksi but are formable at room temperature. They are used for joining both titanium and aluminum sheet. They generally do not need a coating for corrosion protection.

RIVETS

Table 14 - Aluminum and Other Rivet Materials

MATERIAL	RIVET DESIGNATION	RIVET HEADS AVAILABLE	APPLICATIONS
2117-T4	AD	Universal (MS20470) 100° Flush (MS20426	General use for most applications
2024-T4	DD	Universal (MS20470) 100° Flush (MS20426)	Use only as an alternative to 7050-T73 where higher strength is required
1100	A	Universal (MS20470) 100° Flush (MS20426)	Nonstructural
5056-H32	B	Universal (MS20470) 100° Flush (MS20426)	Joints containing magnesium <u>only</u>
Monel (Annealed)	M	Universal (MS20615) 100°Flush (MS20427)	Joining stainless steels, titanium, and Inconel
Copper (Annealed)	---	100° Flush (MS20427)	Nonstructural
7050-T73	E	Universal (MS20470) 100° Flush (MS20426)	Use only where higher strength is required

RIVETS

- **Cherry Buck Rivets (Cherry Rivet Co.)**

  ➛ This rivet is a hybrid, consisting of a factory head and shank of titanium (F_{su} = 95ksi). A shop shank end of ductile titanium/columbium is inertia welded to the pure titanium shank (see Figure 56).

  ➛ This combination allows a shop head to be formed without cracking, while still having a shear strength near 95ksi. These rivets can be used in steel, aluminum or titanium and up to 600°F. They are available in both flush and protruding heads.

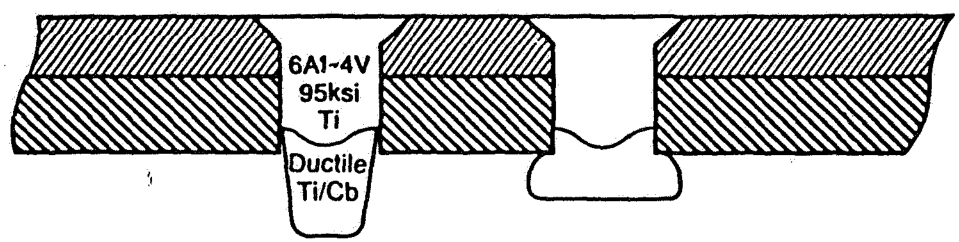


Figure 56 - Cherry Buck Rivet

RIVETS

- ## Cherry Rivets

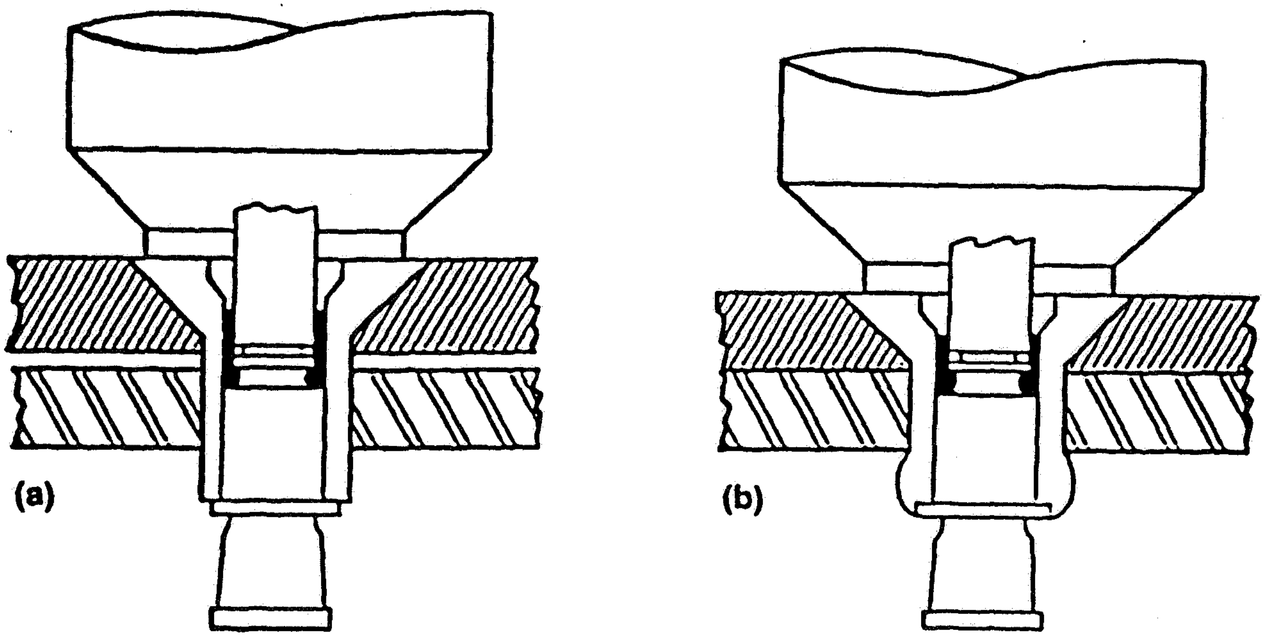
The standard aerospace cherry rivet is blind, with a locking collar, and is installed as shown in Figure 57. They are available in both flush and protruding heads.

Cherry rivets are also available in oversize diameters in the common sizes (1/8 through 1/4 in. diameter) for repairs. The oversize rivet is used where the existing rivet has been drilled out or if a hole diameter is above the accepted tolerance.

These rivets have shear strengths comparable to AD solid aluminum rivets. A typical list of available cherry rivet materials is shown in Table 15.

Note that the use of these blind rivets, along with those of Huck and Allfast, is restricted by the guidelines given in MS33522. Further restrictions by airplane manufacturers limit blind rivets to secondary structures.

RIVETS

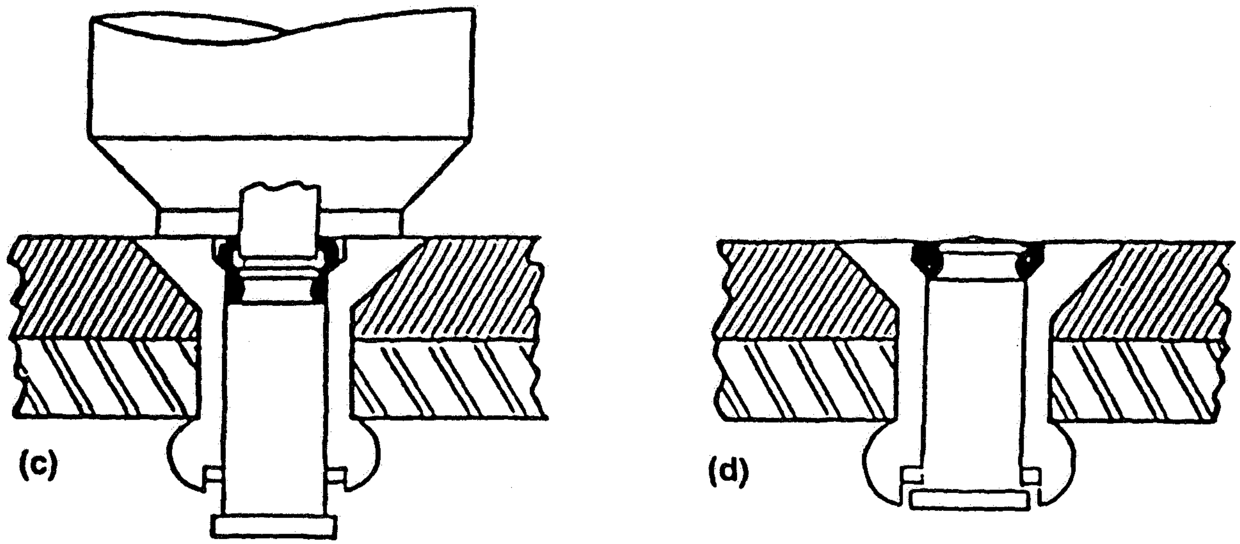


(a) Insert CherryMAX rivet into prepared hole. Place pulling head over rivet stem and apply firm, steady pressure to seat head. Actuate tool.

(b) Stem pulls into rivet sleeve and forms large bulbed blind head; seats rivet head and clamps sheets tightly together. Shank expansion begins.

Figure 57 - Cherry Rivet Installation

RIVETS



(c) "Safe-lock" locking collar moves into rivet sleeve recess. Formation of blind head is completed. Shear-ring has sheared from cone, thereby accommodating a minimum of 1/16 in. in structure thickness variation.

(d) Driving anvil forms "safe-lock" collar into head recess, locking stem and sleeve securely together. Continued pulling fractures stem, providing flush, burr-free, inspectable installation.

Figure 57 (Cont'd) - Cherry Rivet Installation

RIVETS

Table 15 - Cherry Rivet Materials

MATERIALS		ULTIMATE SHEAR STRENGTH, PSI	MAXIMUM TEMPERATURE, °F
SLEEVE	STEM		
5055 Aluminum	Alloy Steel	50 000	250
5056 Aluminum	CRES	50 000	250
Monel	CRES	55 000	900
Inco 600	Inco X750	75 000	1400

RIVETS

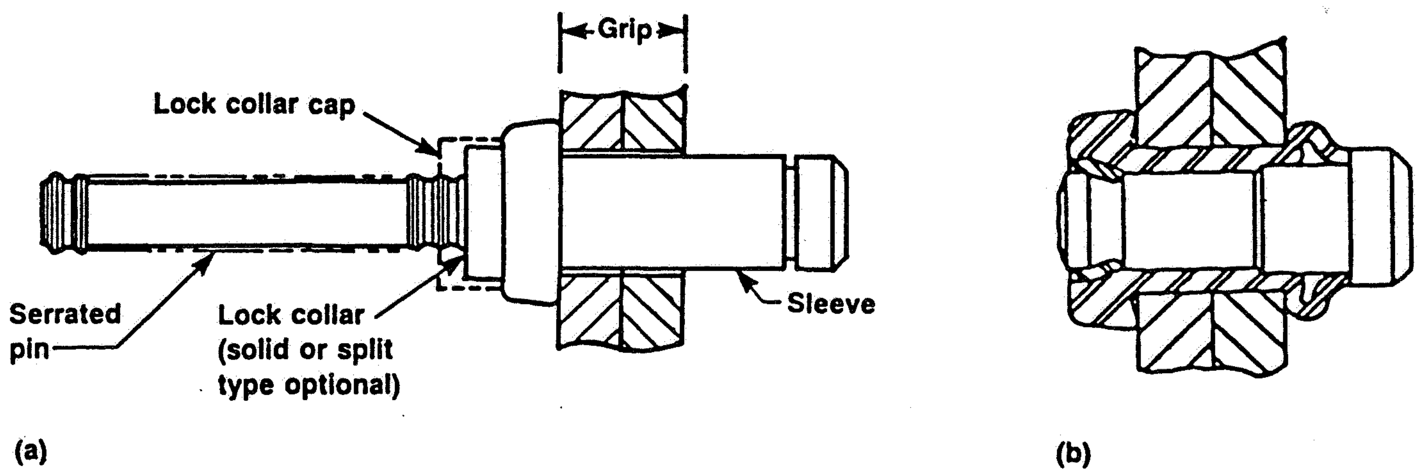
- **Huck Blind Rivets (Huck International, Inc.)**

  These rivets are very similar to cherry rivets, except that in some cases, they are available in higher material strengths. These rivets are made both with and without a locking collar, and in flush or protruding heads.

  A standard Huck rivet is shown in Figure 58. However, the Huck-Clinch rivet has a separate deforming sleeve as shown in Figure 59.

RIVETS

- **Huck Blind Rivets** (Cont'd)



(a) Protruding head, BP-T (MS90354) or BP-EU (MS21141)
(b) Installed fastener

Figure 58 - Huck Blind Rivets

RIVETS

- **Huck Blind Rivets (Cont'd)**

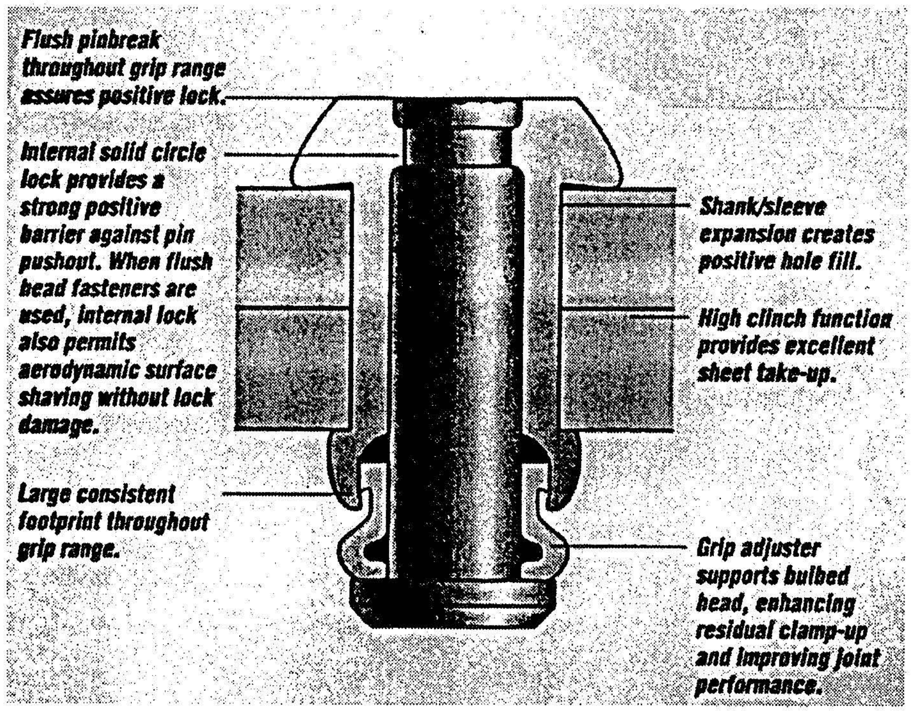


Figure 59 - Huck Clinch Rivet

RIVETS

- **Allfast Blind Rivets (Allfast Rivet Co.)**

  Allfast, like Huck and Cherry, makes several types of rivets, both solid and blind. Their wiredraw rivet (shown in Figure 60) has a tapered stem bulb. As the bulb is pulled through, it expands the tubular body until the tensile load breaks the stem off. The locking collar is pressed in place around the broken stem.

Rivets

- **Allfast Blind Rivets** (Cont'd)

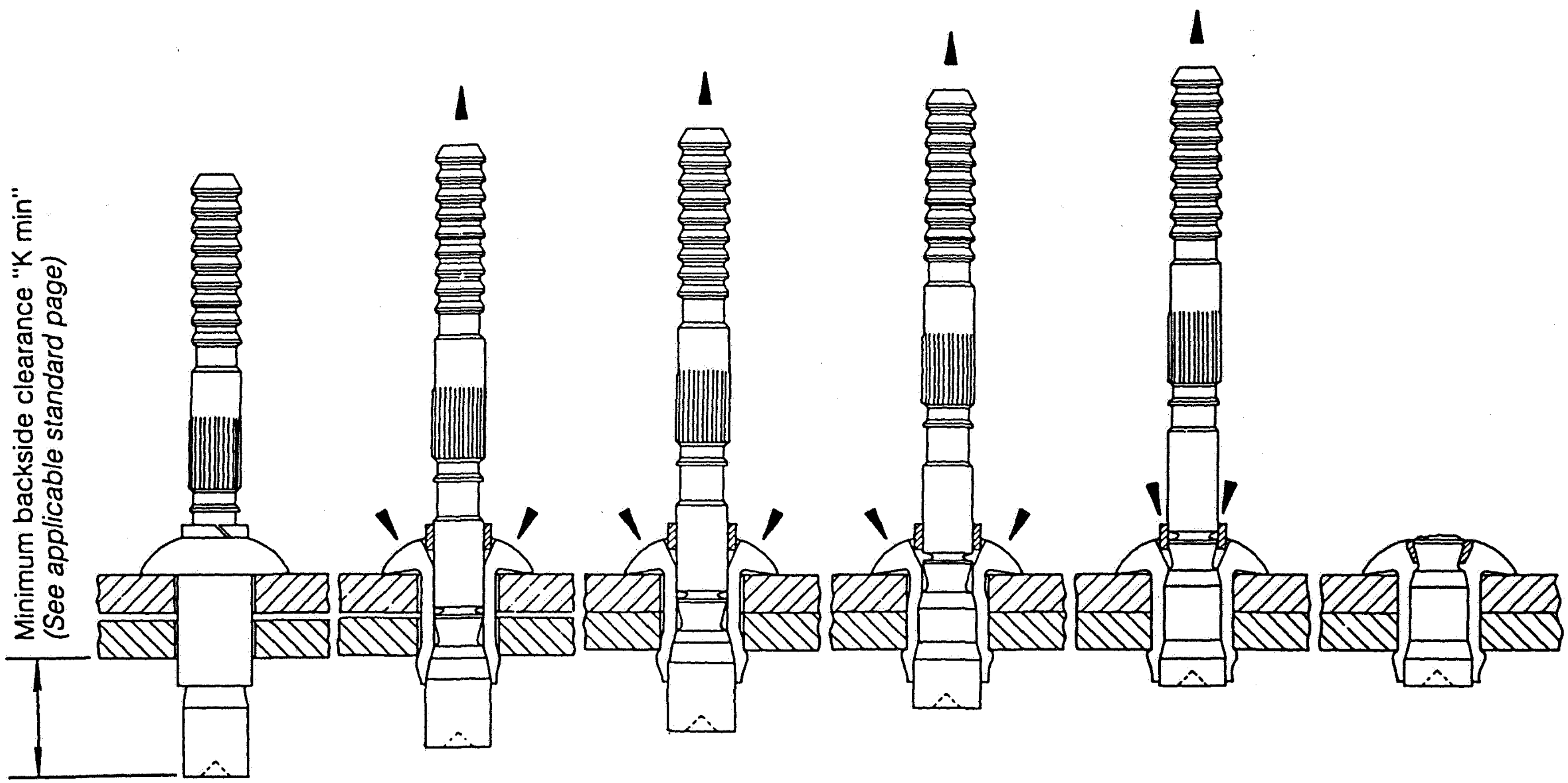


Figure 60 - Allfast Wiredraw Rivet Installation

RIVETS

- ## Hi-Shear Rivets (Hi-Shear Corp.)

Hi-shear rivets consist of a high-strength carbon alloy steel, stainless steel, aluminum or titanium rivet (pin) with a necked-down shop head. This rivet is inserted in a drilled and reamed (not interference fit) hole and a relatively soft collar is swaged in place (as shown in Figure 61). The collar is made of 2024-T4 aluminum or monel.

The Hi-shear rivet head and grip length are sized such that the collar installation is visually inspectable for proper installation.

RIVETS

- ## Hi-Shear Rivets (Cont'd)

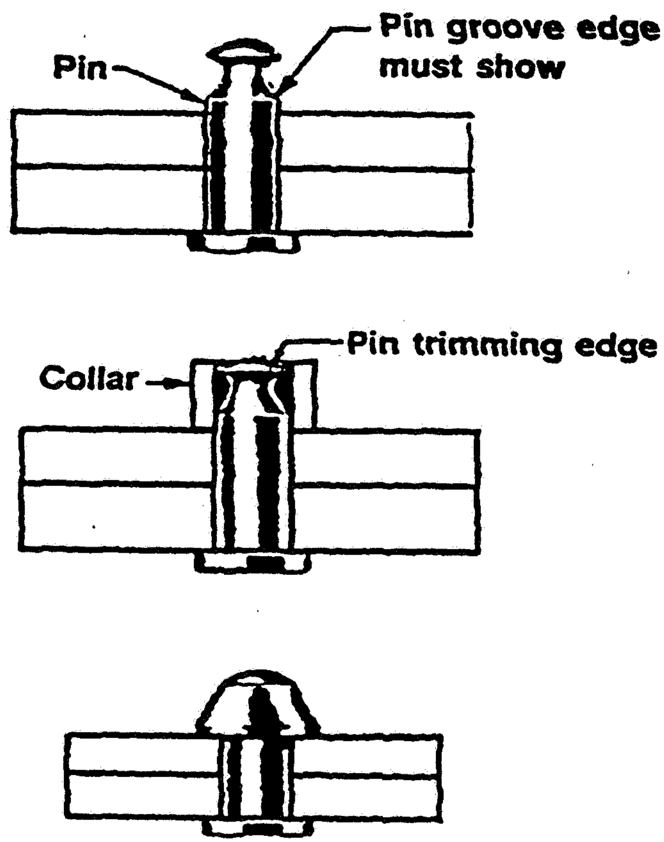


Figure 61 - Hi-Shear Installation

LOCKBOLTS

- ## Introduction

In general, a lockbolt is a non-expanding high strength fastener which has either a swaged collar or a threaded collar to lock it in place. It is installed in a close toleranced drilled hole, but not an interference fit. (An exception, the Taper-Lok, will be covered later.)

A lockbolt is similar to a rivet, in the respect that it is hard to remove once installed and is usually not as strong in tension as a bolt/nut assembly.

Some lockbolts are blind, while others must be fed through from the blind side for near side installation. Lockbolts are available with either countersunk or protruding heads.

Since it is difficult to inspect a lockbolt installation, a bolt/nut assembly should be more reliable. However, lockbolts are faster to install than a bolted assembly.

LOCKBOLTS

- ## Jo-Bolt

Jo-bolts are similar to blind rivets in appearance, but the shank does not expand during installation. The locking collar (sleeve) is expanded, to form a shop head by rotating the threaded stem with a gun while holding the hex head to prevent rotation. The threaded stem is notched and breaks off when the proper torque is reached. (See Figure 62 for a Jo-bolt installation.)

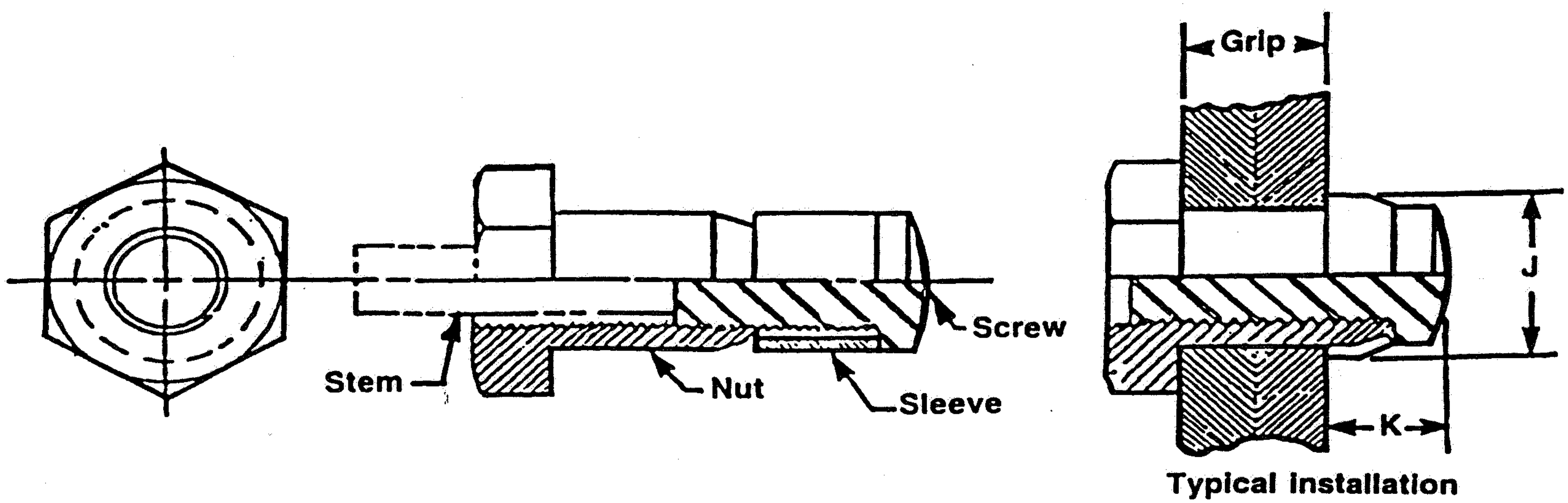


Figure 62 - Jo-Bolt

LOCKBOLTS

- **Huckbolt (Huck Manufacturing Co.)**

A Huckbolt is a lockbolt with serrations on the shank, rather than threads. The collar is swaged in place with a tool which pulls the pin tight during the swaging operation. Then the pin is broken off at the notched area. (See Figure 63 for a Huckbolt installation.)

Since serrations cannot unthread, Huckbolts are used in heavy truck bodies and other areas where both assembly automation and vibration resistance are desired. However, they should not be used in tension, since the collar must be soft to allow swaging.

Huckbolts are available in carbon steel, stainless steel, and aluminum.

LOCKBOLTS

- **Huckbolt** (Cont'd)

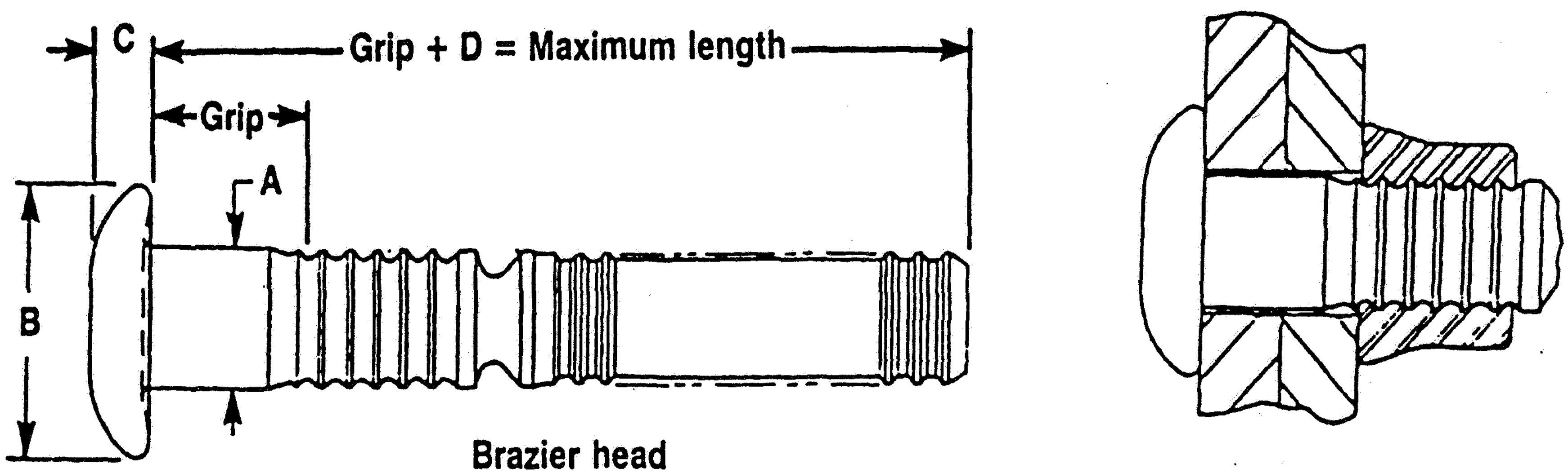


Figure 63 - Installed Huckbolt

LOCKBOLTS

- ## Hi-Lok (Hi-Shear Corporation)

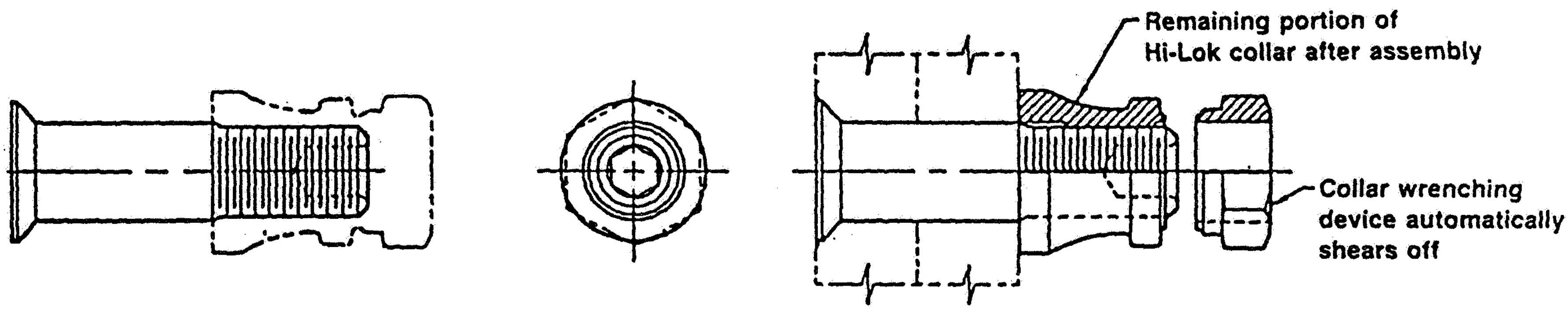
The Hi-Lok lockbolt has either a countersunk or protruding manufactured head and threaded shank like a bolt. It is fed through the hole from the far side. The shank is held with a hexagonal key to prevent rotation while the nut is being torqued with a tool. The outer portion of the nut breaks off at a notch when the proper torque is reached. (See Figure 64 for Hi-Lok installation.)

Hi-Lok lockbolts are available in carbon alloy steel and H-11 tool steel (to 156ksi shear), stainless steel (to 132ksi shear), and titanium (to 95ksi shear).

Note that H-11 tool steel is stress corrosion sensitive and brittle at this strength level. Consequently, some aerospace companies are discouraging its usage as a lockbolt material.

LOCKBOLTS

- ## Hi-Lok (Cont'd)



(a) (b)

(a) Hi-Lok pin.
(b) Hi-Lok pin and collar after assembly.

Figure 64 - Hi-Lok Installation

LOCKBOLTS

- ## Hi-Tigue (Hi-Shear Corporation)

The Hi-Tigue is a Hi-Lok which is driven into an interference fit hole before the collar is installed. This is possible because the Hi-Lok/Hi-Tigue threads have an undersized outside diameter.

The interference fit installation of these fasteners increases the assembly fatigue resistance.

Other than the initial drive-fit of the Hi-Tigue, its installation is the same as a Hi-Lok.

LOCKBOLTS

- Taper-Lok (SPS Technologies)

  A Taper-Lok is a high-strength threaded lockbolt with a tapered shank. It is installed with an interference fit in a drilled and reamed hole. Since the shank taper is only 1.19° and the shank is lubricated, no hole taper is necessary.

  The interference fit prevents shank rotation while the locknut (with a captive washer) is installed. (See Figure 65 for Taper-Lok installation.)

LOCKBOLTS

- **Taper-Lok** (Cont'd)

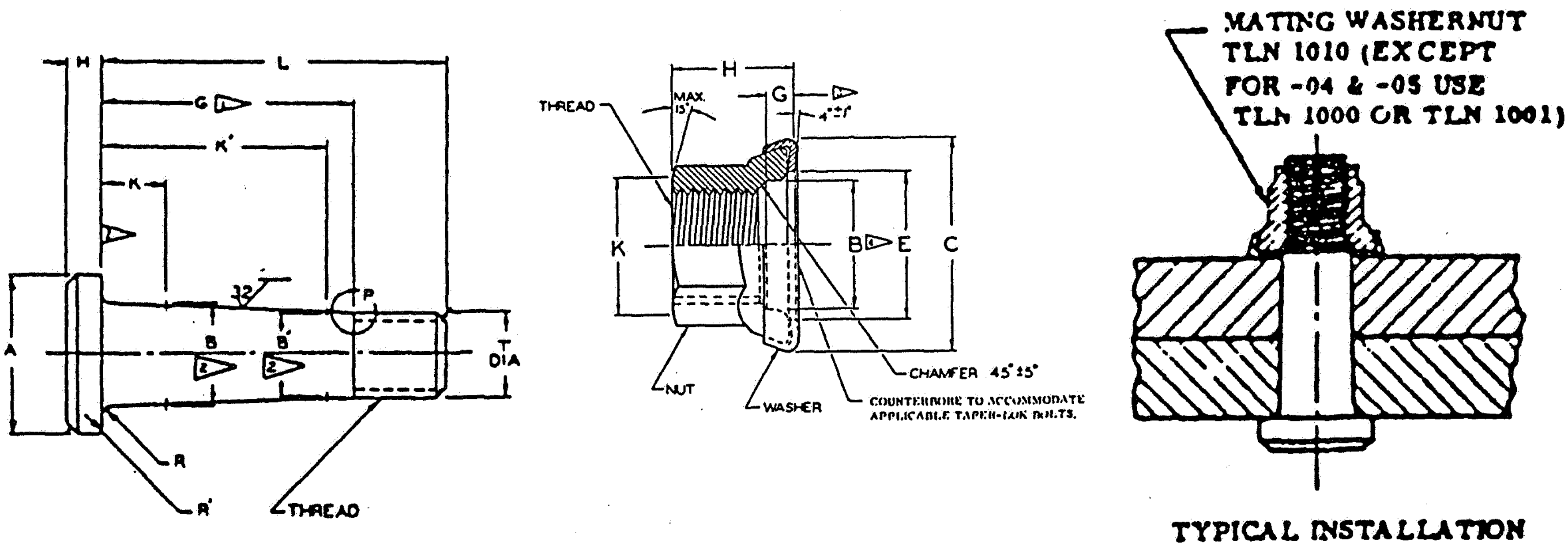


Figure 65 - Taper-Lok Installation

LOCKBOLTS

- **Eddie-Bolt 2 (Voi-Shan Co.)**

  This fastener has special deformed threads (flutes) such that it can be used with a special (3-lobe) nut or a swaged collar. The nut (or collar) is deformed into the 5 flutes during installation to provide positive locking.

  This lockbolt has a hex key socket in the threaded end to prevent shank rotation during nut installation. The nut spins on freely until it bottoms out. Then the special socket turns the nut until the lobes deform into the fluted threads (as shown in Figure 66). Installation is done from one side.

  The swaged collar installation requires a bucking bar on the back side to hold the lockbolt in place while the collar is being swaged (as shown in Figure 67).

LOCKBOLTS

• Eddie-Bolt 2 (Cont'd)

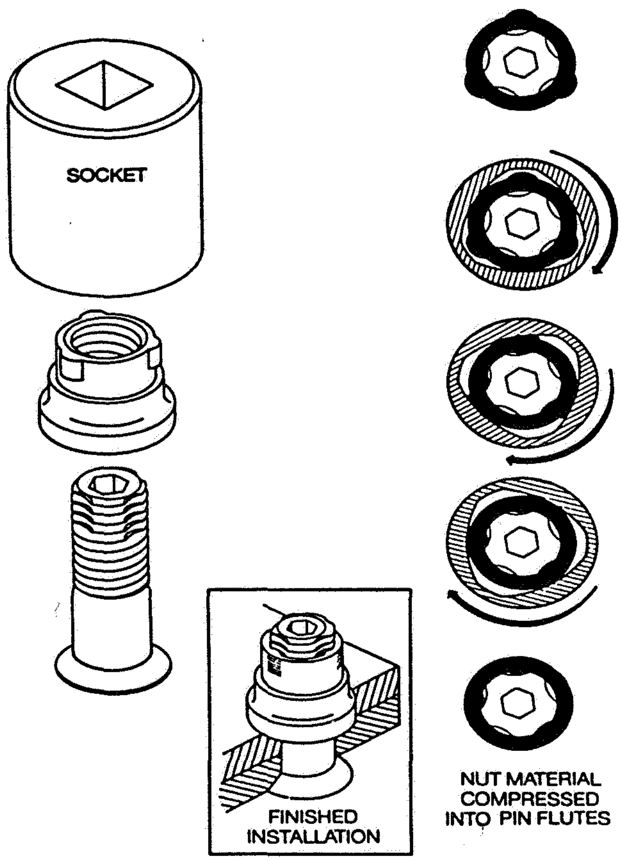


Figure 66 - Threaded Nut

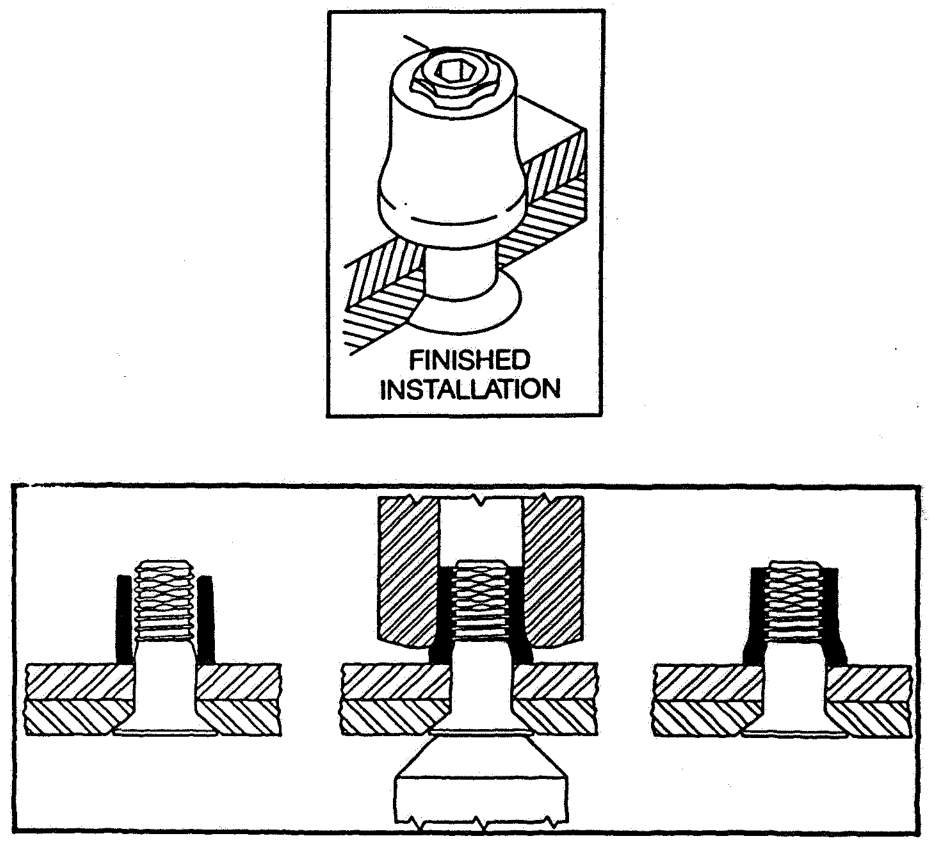


Figure 67 - Swaged Collar

LOCKBOLTS

- **Lightweight Groove Proportioned Lockbolt (LGPL) (by Monogram Aerospace Fasteners)**

This titanium lockbolt is made especially for composite materials. It differs from the standard lockbolt in that it has a 130° countersunk head (vs. 100° or 82°) and a larger diameter collar. These features give smaller contact stresses on the composite surfaces both during and after installation. The collar is 2024-T42 aluminum or commercially pure titanium.

Another feature of the LGPL is a 20° (vs. 30°) flank angle of the serrations to give better collar clamping loads (see Figure 68). Figure 69 shows a LGPL installation.

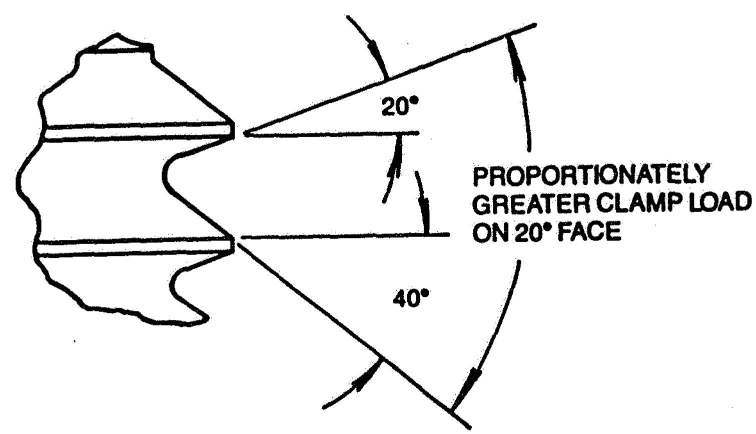
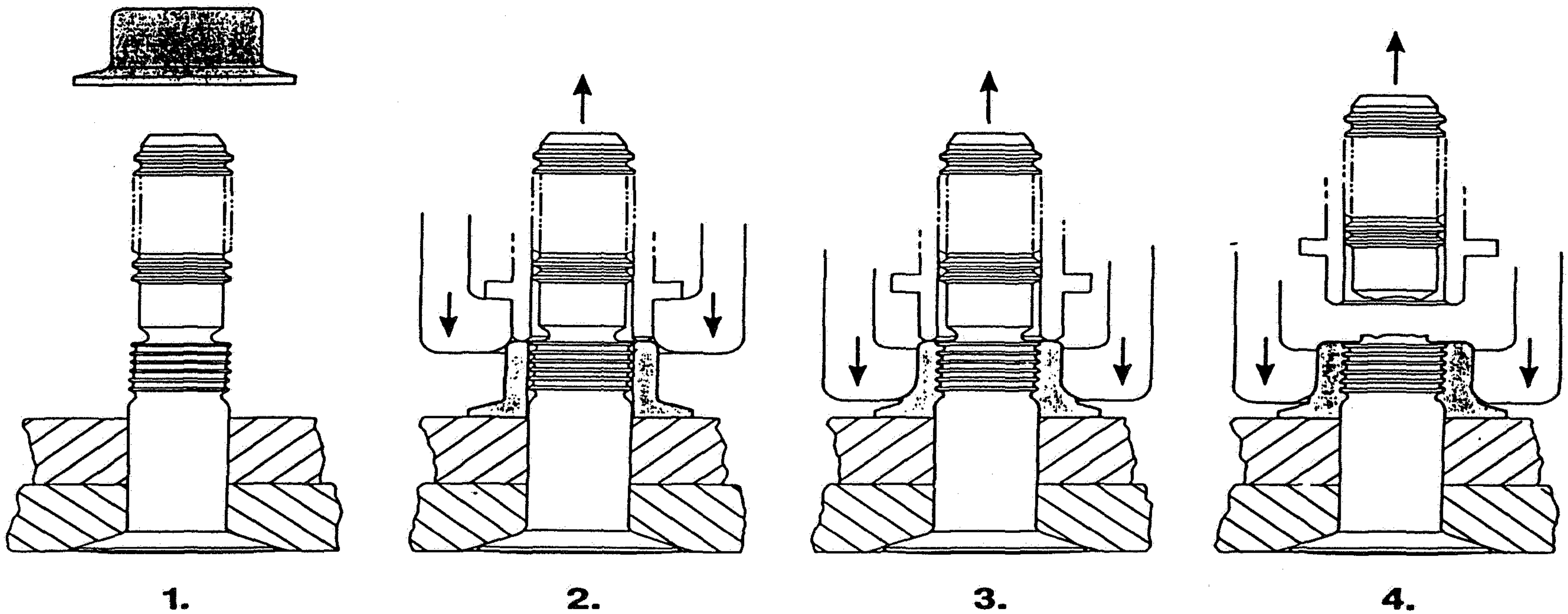


Figure 68
LGPL Flank Angle

LOCKBOLTS

• Lightweight Groove Proportioned Lockbolt (LGPL) (Cont'd)



1. Flanged collar is placed over lightweight pin.
2. Installation tool grips and pulls pin, drawing sheets tightly together and removing sheet gap.
3. As the pull on the pin increases, the tool anvil swages the flanged collar into the locking grooves forming a permanent vibration resistant lock.
4. Pull on pin continues until pin fractures at the breakneck groove and is then ejected. Tool anvil disengages swaged collar.

Figure 69 - LGPL Installation

LOCKBOLTS

- **General Guidelines for Selecting Rivets and Lockbolts**
  - ➛ Don't use expanding rivets in composites.
  - ➛ Don't use 5056 aluminum rivets in anything other than magnesium, since 5056 is stress corrosion sensitive.
  - ➛ A threaded lockbolt can carry up to the tensile allowable of the shank, but each design should be checked individually.
  - ➛ Since drilled fastener holes are not plated or coated, it is necessary to coat the material surfaces with an insulator/sealer paste or paint to retard galvanic corrosion between the fastener and the joint material.
  - ➛ MIL-HDBK-5 gives rivet design allowables which are based on test data.

LOCKBOLTS

- **General Guidelines for Selecting Rivets and Lockbolts** (Cont'd)

  ➣ Rivet installation is covered by MIL-STD-403.

  ➣ Some corrosion prevention methods are covered by MIL-P-116 and MIL-STD-171.

  ➣ Design and selection requirements for blind *STRUCTURAL* rivets are given in MS33522.

  ➣ Testing of fasteners is covered in MIL-STD-1312.

  ➣ NAS523 gives rivet codes and callouts.

  ➣ Review the fastener manufacturer's design criteria before incorporating the fasteners into your design.

INSPECTION AND ACCEPTANCE OF FASTENERS

- Introduction

We can specify what we want in a fastener for our design, but we need acceptance/inspection criteria and procedures to show that we got what we ordered. The criticality of the fastener should determine the amount of inspection/testing required.

A variety of inspection methods will be covered in subsequent pages.

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Fastener Hardness Testing

  - The hardness of a material can be equated to its tensile strength. Hardness is measured by the resistance of a material to indentation by a ball or a plain or truncated cone. Brinell and Rockwell are the most common methods, but Vickers and Knoop are also used.

  - A Brinell test is performed by pressing a hardened steel ball into a test specimen.

  - A Rockwell test is similar to a Brinell test, except that the loads are smaller, and hence the resulting indentation is smaller and shallower.

  - A Rockwell test is applicable to testing of materials with hardnesses above the range of the Brinell test.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Fastener Hardness Testing** (Cont'd)

  - ➢ A Rockwell test uses a B scale for medium hardness materials and a C scale for those harder than 100 on the B scale (B100).

  - ➢ Figure 70 shows a Brinell hardness tester and Figure 71 shows a Rockwell hardness tester.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Fastener Hardness Testing** (Cont'd)

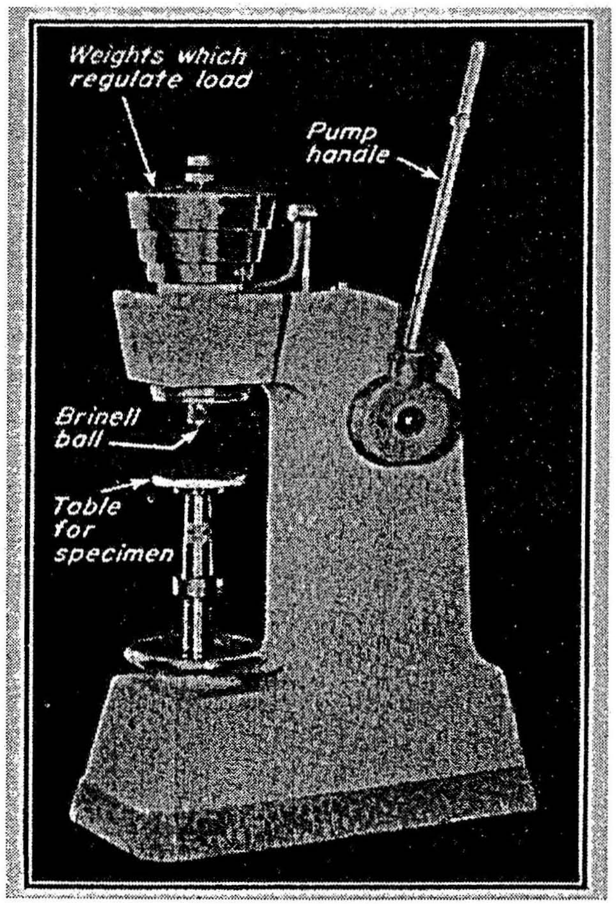


Figure 70 - Brinell Hardness Tester

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Fastener Hardness Testing (Cont'd)**

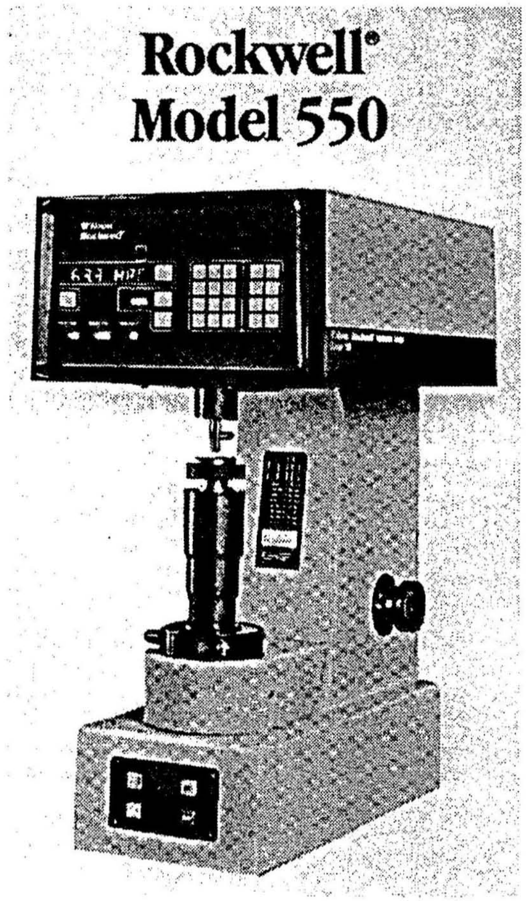


Figure 71 - Rockwell Hardness Tester

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Fastener Hardness Testing (Cont'd)

  - MIL-STD-1312-6 gives a standard test method and specifies apparatus to be used for hardness testing of all types of structural fasteners.

    - The Rockwell Hardness Tester has a B scale with a 1/16-inch diameter brale penetrator and a major load of 100 kilograms. The C scale has a 120 degree conical brale with a 0.2 millimeter tip radius and a major load of 150 kilograms.

    - A Rockwell Superficial Hardness Tester is similar to the normal Rockwell Hardness Tester except that the indentation is much shallower.

    - A Knoop or Vickers Micro Hardness Tester is used when the size or configuration of the test piece does not permit testing on the normal or superficial Rockwell equipment.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Fastener Hardness Testing (Cont'd)**
  - ➛ Fastener specimens consist of flat or mounted specimens.
    - ▸ Flat specimens are machined and ground to obtain two flat, parallel surfaces, one of which is used for hardness readings.
    - ▸ Mounted specimens are used for fasteners that, because of size or configuration, cannot be machined with two parallel surfaces. They are imbedded in a thermosetting or thermoplastic material and prepared by sanding or polishing.
  - ➛ In general, #0 to #5 size bolts, screws, studs, and nuts, 0.0625 to 0.125 inch diameter rivets, and all socket set screws are mounted. Most larger sizes are ground flat.
  - ➛ On larger diameter fasteners, it is possible to get a rough hardness reading on the unthreaded shank. However, this reading will be high if the fastener has been cold-formed.

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Tensile Test

Tensile testing of samples is sometimes done as an acceptance procedure.

The criticality of the fasteners determines the test sample size for the lot.

A standard tensile testing machine (with large fixtures) is used in order to minimize fixture deformation.

Both tensile yield and ultimate strengths, as well as elongation, can be determined from this test.

Failure of any test fastener to meet minimum requirements normally rejects the entire lot.

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Carbon Content Test

One of the favorite ways to counterfeit medium carbon fasteners is to lower the carbon content and add boron to increase hardening capability. For this reason a carbon content acceptance test is run to determine the amount of carbon and/or sulfur in the fastener materials.

- High-temperature combustion is used for this test.
- Two types of furnaces may be used:
  - High frequency
  - Resistance high-temperature
- Two methods of carbon/sulfur detection are used:
  - Infrared absorption
  - Thermal conductivity

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Carbon Content Test** (Cont'd)

  ➣ **Test Theory**

  To determine the content of carbon and sulfur in a material, the principle of high-temperature combustion with oxygen injection is used. This causes carbon and sulfur to form carbon dioxide (CO_2) and sulfur dioxide (SO_2).

  The combination of a high-temperature furnace capable of 1370-1425 °C (2500-2600 °F) and a chamber flooded with oxygen causes the sample to burn.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Carbon Content Test (Cont'd)**

  ➛ **Test Theory (Cont'd)**

  During combustion, other oxide compounds form and are separated to ensure detection of only the desired elements.

  Metal oxides are normally removed by using a porous trap that allows gasses to pass through while stopping the metal oxide particles.

  Moisture is removed with a desiccant such as magnesium perchlorate.

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Carbon Content Test (Cont'd)

  - Samples
    - *Form*: Solids, chips, or powders
    - *Size*: 1 gram or less, depending on the type of material
    - *Preparation*: Bulk samples must be cut to prescribed size required for determination. Specimens should not be contaminated with carbon or sulfur before analysis.
  - Limitations
    - Specimen must be homogeneous.
    - Graphite-bearing specimens require special handling.
    - Method is destructive.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Carbon Content Test (Cont'd)**

  - Estimated Analysis Time

    - *Sample Preparation*: 2 to 3 minutes.
    - *Analysis*: 40 seconds to 2 minutes.

  - High-Frequency Furnace (See Figure 72)

    The central part of a high-frequency furnace is the load or work coil section.

    Some portion of the ceramic crucible which holds the sample must be inductive; if sample is not, inductive accelerators are required.

    High-frequency power to the work coil establishes a high-frequency field around the sample. This field couples with the inductive material in the crucible and heats the specimen.

INSPECTION AND ACCEPTANCE OF FASTENERS

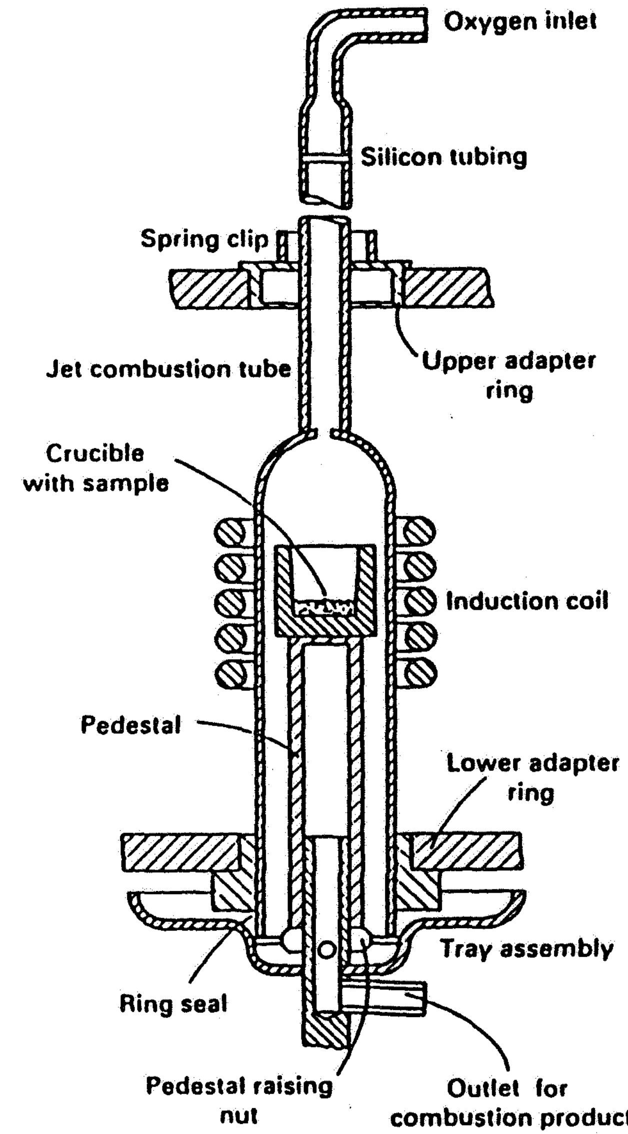


Figure 72 - Typical High-Frequency Combustion Configuration

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Carbon Content Test (Cont'd)**

  ➛ **Resistance High-Temperature Furnace**

  Carbide or molydisilicide elements are used to heat the furnace.

  The coupling is unimportant, but the combustion point of the sample is significant.

  Accelerators used with these furnaces will normally combust more readily than the specimens.

  Detection of the separated gasses is most commonly provided by one of two detection systems that provide a specific and consistent measurable signal. This signal is processed electronically to provide a percentage of each element by weight.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Carbon Content Test (Cont'd)**

  - Infrared Detection

    Applied on the basis that various gases can absorb energy within a specific wavelength of the infrared spectrum.

    Amount of energy absorbed by combustion gases within the CO_2 and SO_2 absorption wavelengths determines the carbon and/or sulfur content.

  - Thermal-Conductive Detection System

    This system is based on the principle that each gas has a distinct capability of carrying heat from a body.

    It responds to changes in heat conduction.

    The thermal-conductive change generated by CO_2 extracted from the sample is proportional to the amount of carbon extracted.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Inspection**

The common methods of inspection (per MIL-HDBK-H28/20B) are Systems 21, 22, and 23. (These Systems are also covered in ANSI B1.3 M.)

In general, System 21 requires the least amount of inspection, System 22 requires an intermediate level, and System 23 is the most stringent.

Since a complete treatise of each method would be too lengthy, a summary of each method will be given here. (See Figures 5(a) and 5(b) for thread terminology.)

INSPECTION AND ACCEPTANCE OF FASTENERS

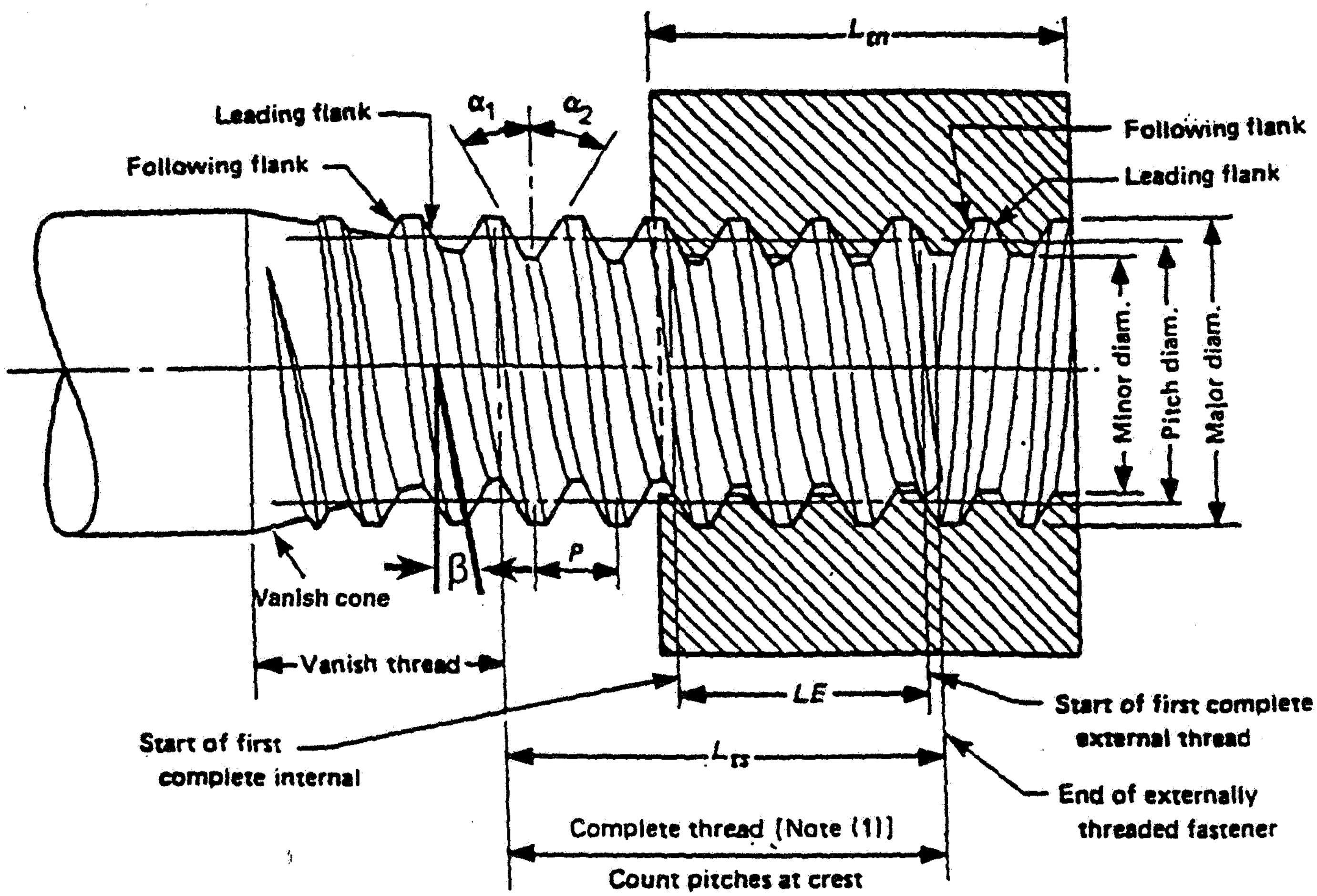


Figure 5(a) - Thread Terminology

INSPECTION AND ACCEPTANCE OF FASTENERS

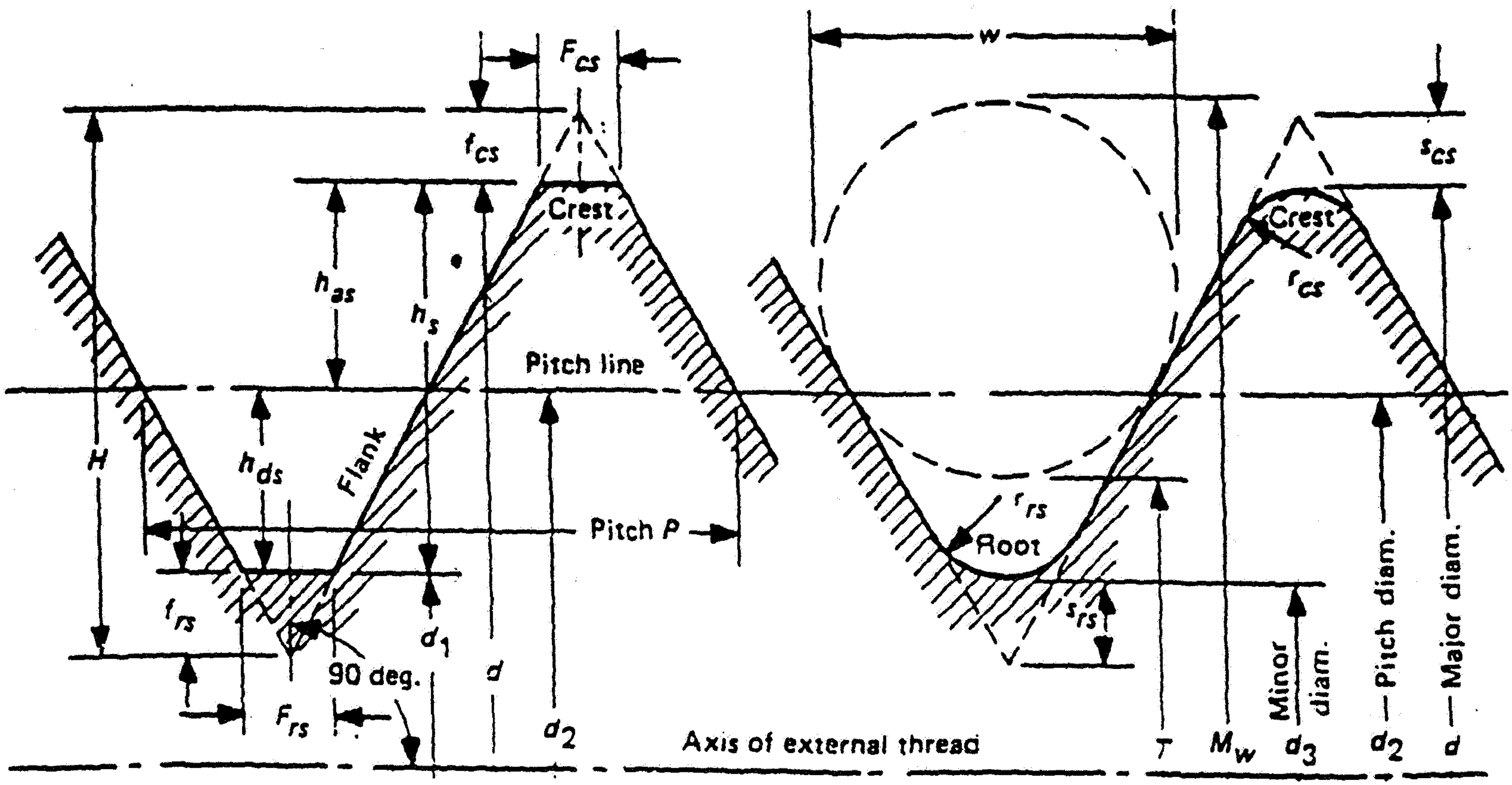


Figure 5(b) - External Thread

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Inspection** (Cont'd)
  - External Threads
    - System 21 includes:
      - Go-No Go functional diameter
      - Major diameter
    - System 22 includes System 21 measurements PLUS:
      - Pitch Diameter (Measured with pitch micrometer or 3-pins)
      - Thread groove diameter
      - Functional diameter
      - Lead and flank angles (Go-No Go)
      - Minor diameter
      - Root Profile (radius)

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Inspection** (Cont'd)
  - External Threads (Cont'd)
    - System 23 includes Systems 21 and 22 PLUS:
      - Roundness of pitch cylinder
      - Taper of pitch cylinder
      - Cumulative thread form variation
      - Lead and helix angle variation
      - Flank angle variation
      - Runout variation
      - Surface texture

NOTE: Visual inspection for thread manufacturing defects must *STILL* be done.

INSPECTION AND ACCEPTANCE OF FASTENERS

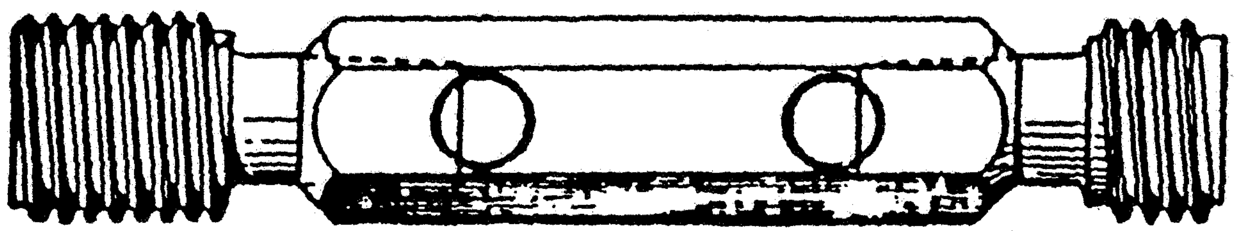
- **Thread Inspection (Cont'd)**
  - Internal Threads
    - System 21 includes:
      - Go - No Go functional diameter (See Figure 73)
      - Minor diameter (See Figure 74)
    - System 22 includes System 21 measurements PLUS:
      - Minimum material pitch diameter or thread groove diameter
      - Lead (including helix) and flank angles

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Inspection (Cont'd)**

  - Internal Threads (Cont'd)

    - Double ended thread plug limit gage



GO GAGE

NO-GO GAGE

(a) Used to check internal threaded or tapped holes.
(b) Threads are acceptable if Go Gage can and No-Go Gage cannot be turned into the threaded hole.

Figure 73 - Internal Thread Functional Diameter Gage

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Inspection (Cont'd)**

  ➛ Internal Threads (Cont'd)

Gages and Gaging for MJ Series Metric Screw Threads

ANSI/ASME B1.22M-1985 An American National Standard

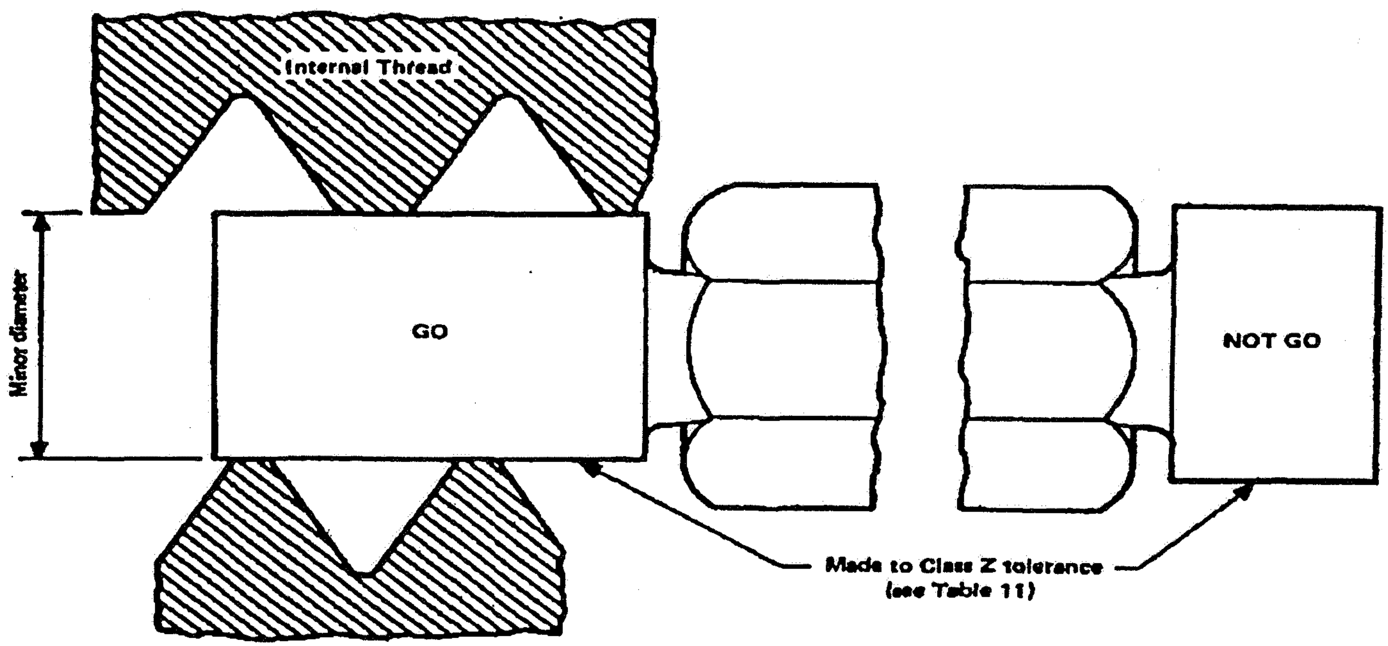


Figure 74 - Internal Thread Minor Diameter Gage

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Inspection (Cont'd)**
  - ➛ **Internal Threads (Cont'd)**
    - ▸ **System 23 includes System 22 measurements *PLUS:***
      - Roundness of pitch cylinder
      - Taper of pitch cylinder

      NOTE: Internal thread root radius is *NOT MEASURED*.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Defects**

  Although Systems 21, 22, and 23 provide for extensive geometric thread inspection, they do not provide for manufacturing defect detection. For defects in the threads, FF-S-86 gives examples of acceptable and unacceptable defects. See Figures 75, 76, and 77.

  Note that the acceptance of thread defects becomes more critical as the fastener strength increases and its ductility decreases.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Defects (Cont'd)**

  ➛ **Threads shall have no laps or seams at the root or on the flanks below the pitch diameter line.**

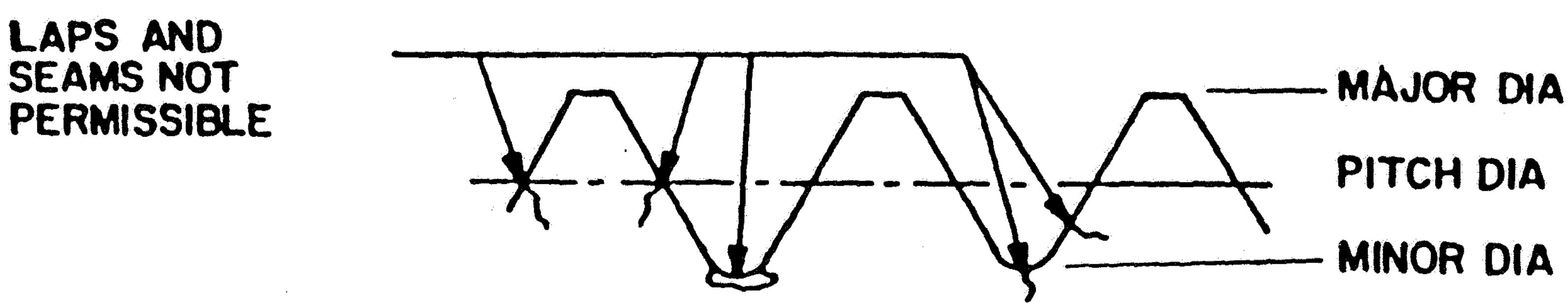


Figure 75 - Discontinuities Below Pitch Diameter Line

Inspection and Acceptance of Fasteners

- **Thread Defects** (Cont'd)

  - ➛ **Laps are permissible at the crest to a depth of 25 percent of the basic thread height and on the flanks above the pitch diameter line.**

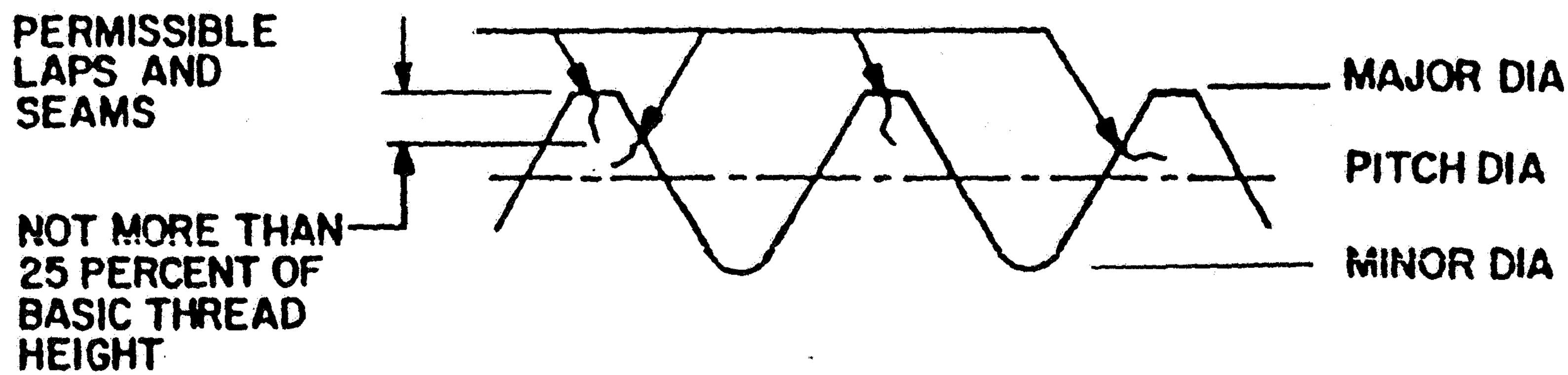


Figure 76 - Discontinuities Above Pitch Diameter Line

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Defects** (Cont'd)

  ➛ **Slight deviations from the thread contour are permissible at the crest of the threads.**

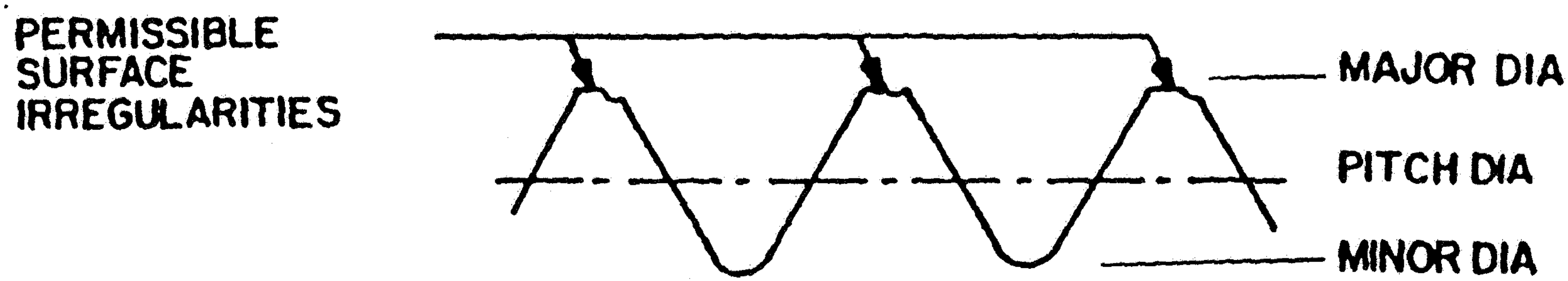


Figure 77 - Discontinuities In Crest Contours

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Thread Non-contact (LASER) Inspection**

A laser-based instrument has been developed to make dimensional measurements on product screw threads. (See Figure 78.)

The developed measurement system uses a laser triangulation sensor, coupled to a precision motion system, to digitize the thread form of a part.

Measurements are made by comparing data obtained by laser scanning to a perfect mating part that has been mathematically created in software. (See Figure 79.)

The screw thread or spindle axis serves as the common reference frame for the scans, allowing profiles taken at various angles of rotation to be related to one another.

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Laser Thread Measurement System

  - Laser thread measurement applications

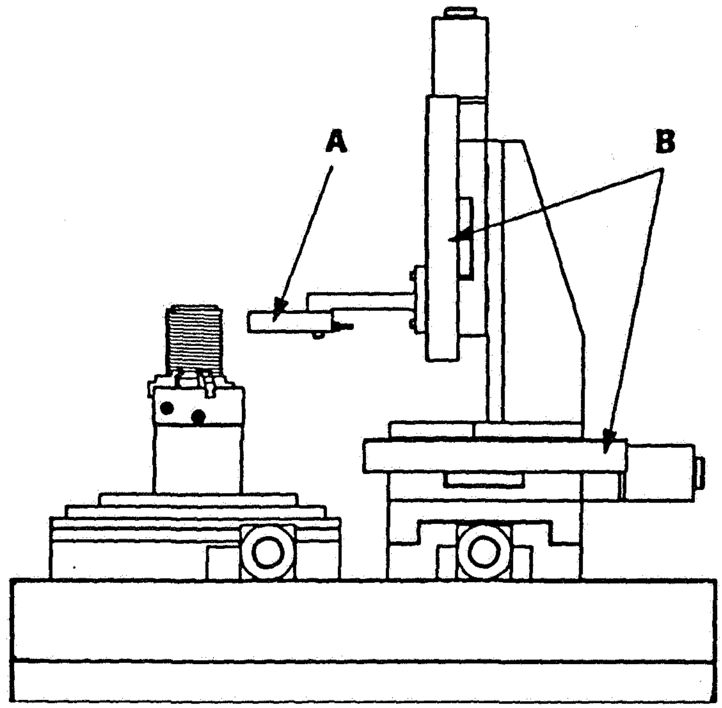
    - To inspect thread plug gages used for tapped hole inspection
    - High precision inspection of externally threaded parts and fasteners where thread strength and fit accuracy are mandatory requirements
    - To inspect dies and taps
    - Can handle parts up to 6 inches in diameter and 4 to 64 threads per inch

  - Laser thread measurement is too time consuming for production lot inspection. It is used when it is absolutely necessary to precisely inspect thread characteristics that are essential for satisfactory performance.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Laser Thread Measurement System (Apeiron, Inc., Bloomington, MN)**

Figure 78



1. Threaded part to be measured is mounted on a rotary fixture located on a precision air bearing spindle.
2. Laser sensor (A) scans threaded part along path parallel to part axis.
3. Glass-scale, linear encoders (B) ensure accurate positioning.
4. Once the laser head scans the threaded part, the rotary table indexes successively to allow additional scans at various angular locations.

INSPECTION AND ACCEPTANCE OF FASTENERS

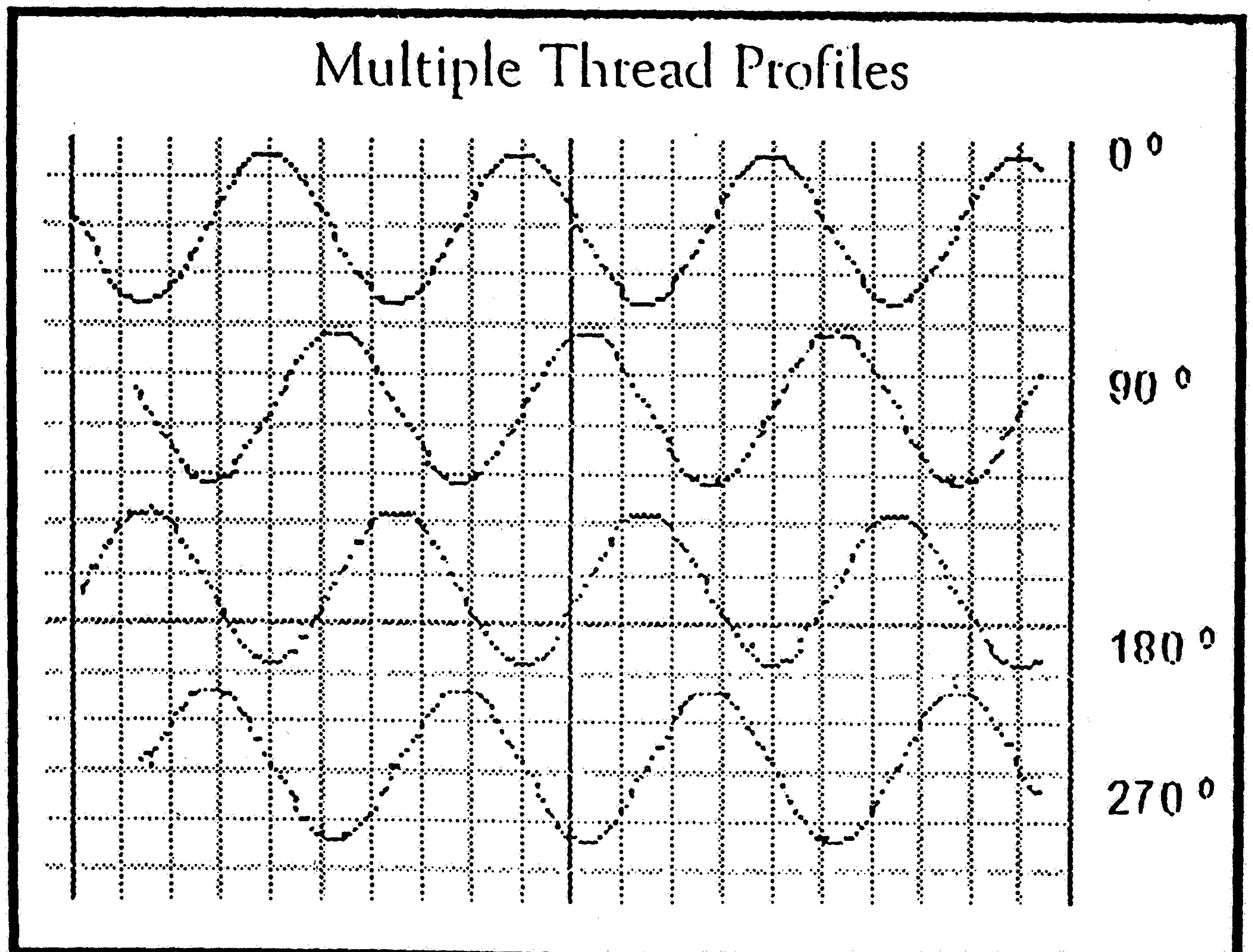


Figure 79 - Multiple Laser Measured Thread Profiles

Inspection and Acceptance of Fasteners

- **Variation in Pitch Diameter**

The effect of pitch diameter variation on thread strength and a fastener's installed joint performance has been a topic of debate for many years.

The Industrial Fasteners Institute (IFI) in Cleveland, Ohio initiated a research effort in 1993 to manufacture, measure and test fasteners deliberately made out of tolerance to develop data that would relate thread pitch diameter variations to fastener performance. (See *Mechanical Engineering, December, 1996*.)

It was concluded that variations in pitch diameter have a very small impact on joint strength, fatigue life, and joint clamping performance. The poorest-fit bolt/nut pitch diameter combination still passed the standard tensile and proof load requirements.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection**

Surface defects in the unthreaded areas must also be inspected and evaluated. The fastener strength and application will have a substantial influence on the acceptance of defects.

A list of defects and their definitions is given in ASTM F788. However, this specification is for fasteners of .25 inch diameter (and up) with tensile strengths of 90 KSI (and up).

The following pages give some of the ASTM F788 acceptance/rejection criteria. However, engineering judgement can still be used to modify these requirements.

Nut inspection is very similar to bolt inspection, so it will not be covered here. See ASTM F812 for nut inspection criteria.

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Head and Shank Inspection (Cont'd)

  - ### Quench Cracks

    Quench cracks are caused by excessively high thermal and transformation stresses during heat treatment. (See Figure 80.)

    Quench cracks of any depth, any length, or any location are not permitted, and are cause for rejection.

  - ### Socket Head Depth

    Although the depth of socket and the depth of remaining head material is given in ANSI B18.3, NAS, and MS specifications, these dimensions are *NOT* checked during incoming inspection. As a result, heads will sometimes break off, due to the socket depth being too deep.

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Head and Shank Inspection (Cont'd)

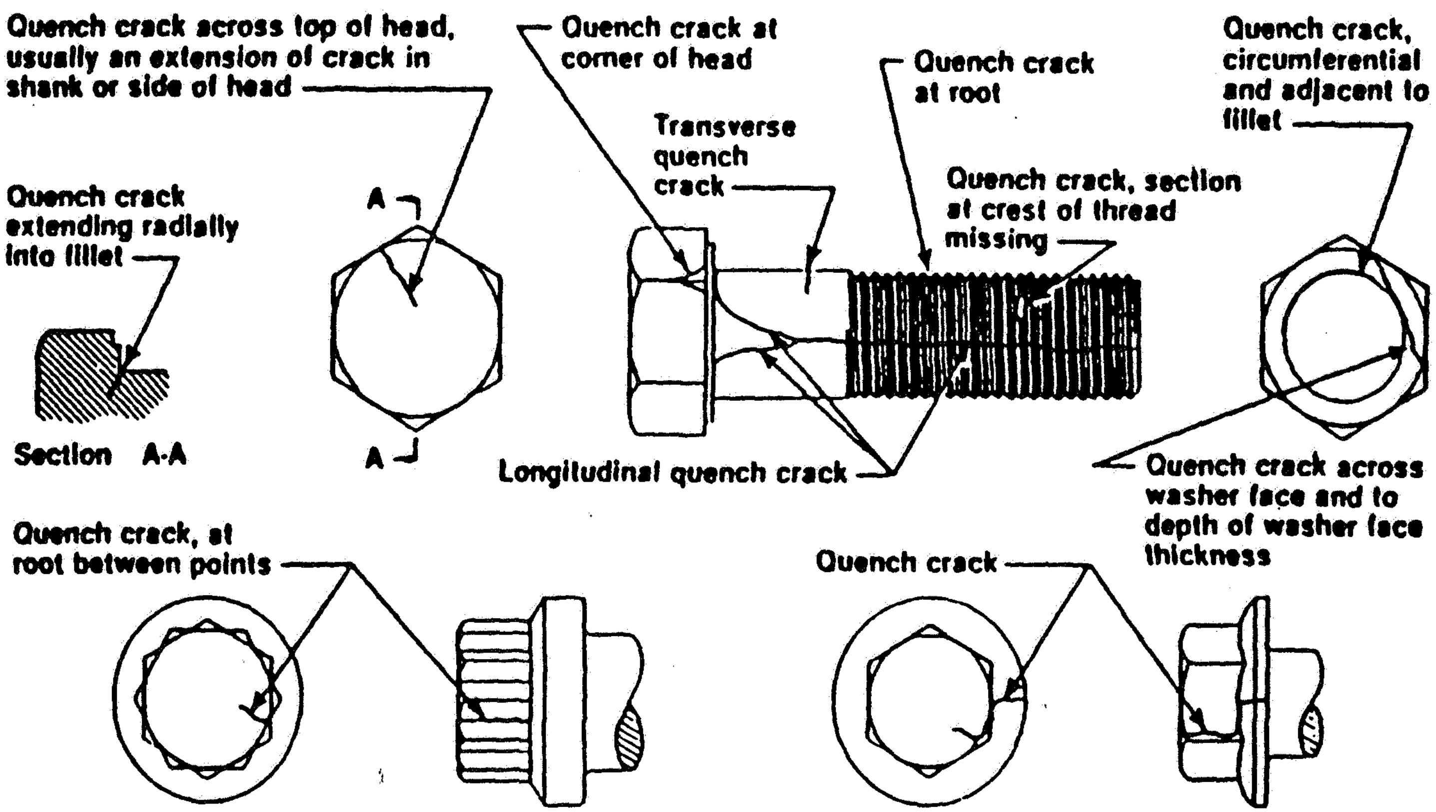


Figure 80 - Typical Quench Cracks

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection (Cont'd)**

  ➛ **Forging Cracks**

  Forging cracks may occur during the cutoff or forging operation. They are located on the top of the head and on the raised periphery of indented head bolts and screws.

  Forging cracks on the top of bolts and screws are permitted *ONLY IF:*

  - No crack shall have a length exceeding 1.0 x the basic major thread diameter.
  - No crack shall have a width or depth exceeding 0.04 x the basic major thread diameter. (See Figure 81.)

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection** (Cont'd)

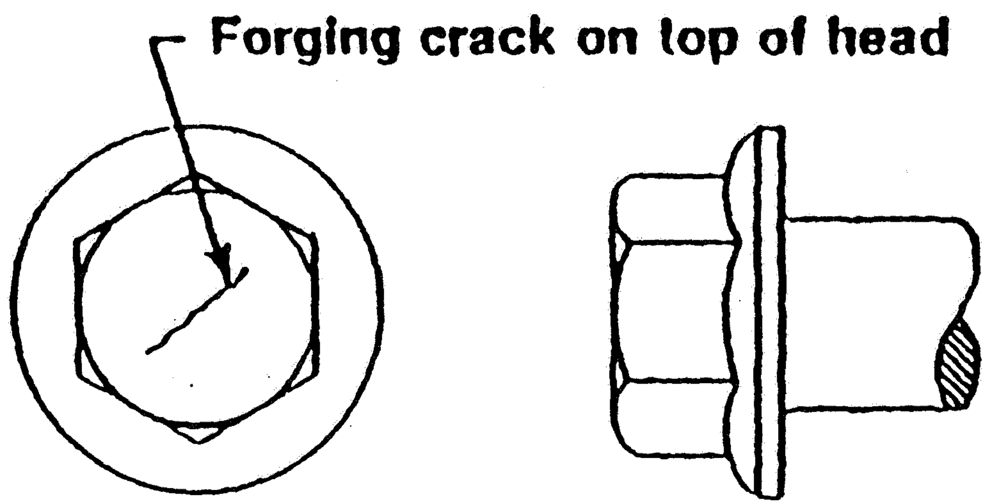


Figure 81 - Typical Forging Cracks

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Head and Shank Inspection (Cont'd)

  - Bursts and Shear Bursts

    A burst is an open break in the metal and a shear burst is an open break in the metal located at approximately 45 deg. to the product axis. (See Figure 82.)

    For hex-head bolts and screws, bursts and shear bursts are permitted *ONLY IF:*

    - No burst or shear burst in the flats extends into the crown (chamfer circle) on the top of the head or into the underhead bearing circle.

    - No burst or shear burst located at the intersection of the wrenching flats reduces the width across corners below its specified minimum.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection (Cont'd)**
  - ➛ Bursts and Shear Bursts (Cont'd)

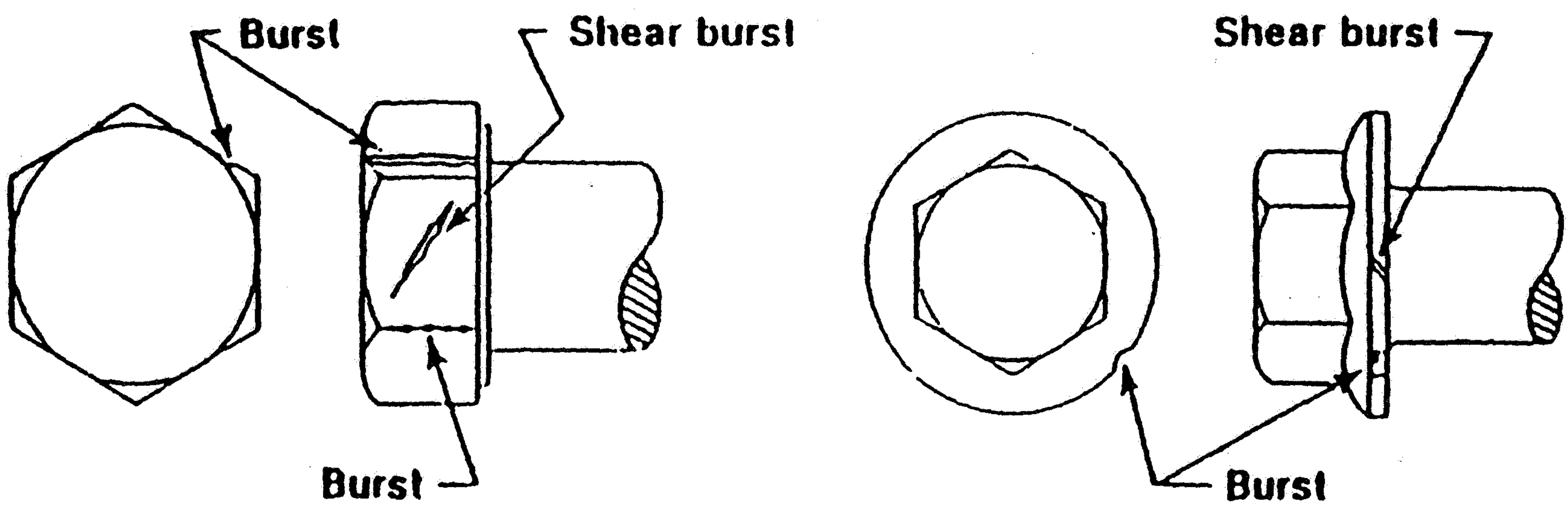


Figure 82 - Typical Bursts and Shear Bursts

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection (Cont'd)**

  ➛ **Folds**

  A fold is a doubling over of metal which occurs during the forging operation and may occur at or near the intersection of diameter changes. They are especially prevalent with noncircular heads. (See Figure 83.)

  Folds at interior corners below the underhead bearing surface (junction of head to shank) are not permitted.

  Folds at interior corners above the underhead bearing surface (junction of hex head to shank) are permitted.

  Folds at exterior corners are permitted.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection (Cont'd)**

  ➣ **Folds (Cont'd)**

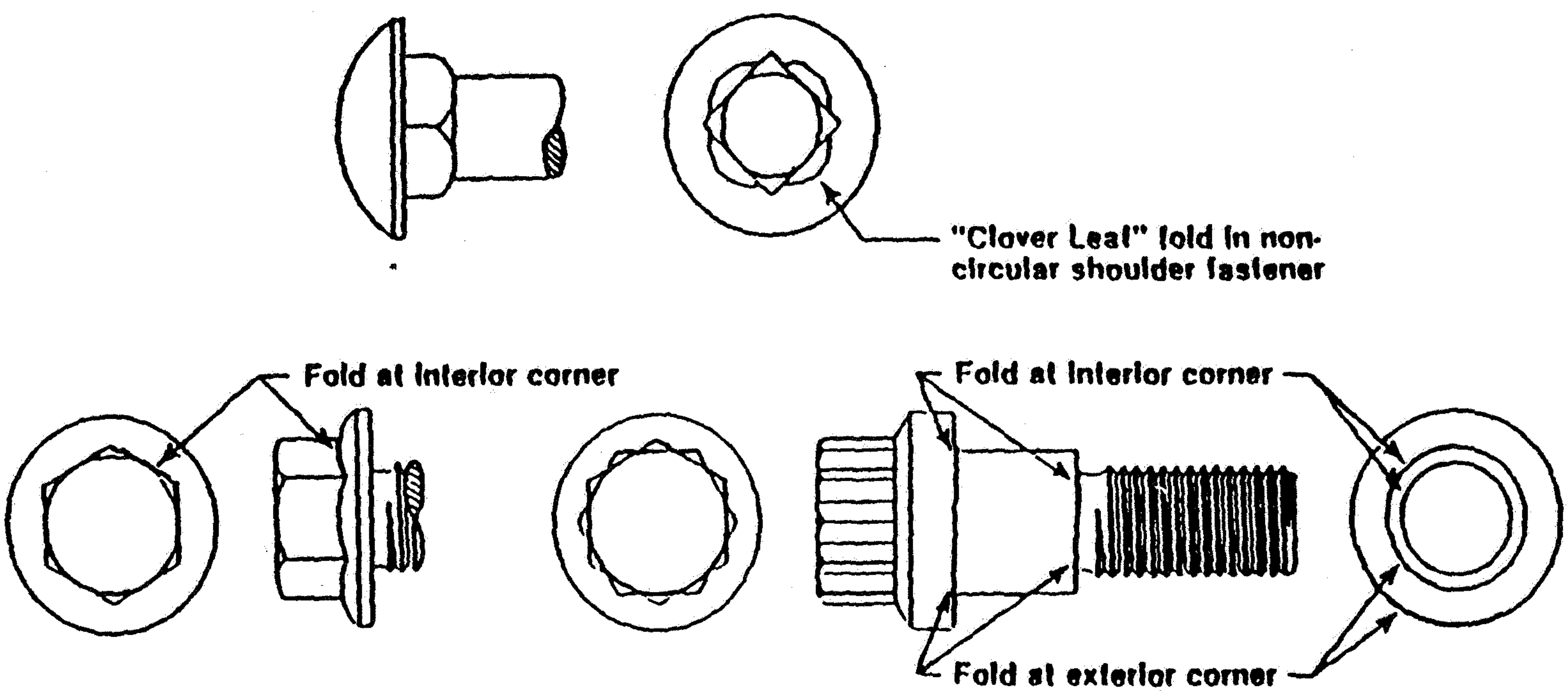


Figure 83 - Typical Folds

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection (Cont'd)**

  ➛ **Seams**

  Seams are usually in the raw material and are normally straight or smooth-curved lines.

  Seams are permitted in shanks if their depth is less than 0.03X the basic major thread diameter.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection (Cont'd)**

  - **Surface Voids**

    A surface void is a shallow pocket or hollow due to non-filling of metal during forming.

    Voids are permissible if:

    - Void depth does not exceed 0.010 inch or 0.02X the shank diameter (whichever is greatest).

    - Combined void areas on the underhead bearing area is less than 10% of the minimum bearing area (subject to engineering approval).

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Head and Shank Inspection** (Cont'd)

  → Tool Marks, Nicks & Gouges

  Tool marks, nicks and gouges are permitted on the underhead bearing surface if they still allow a surface roughness under 125 micro-inches.

  In other areas, the acceptance of these marks is an engineering judgement call.

INSPECTION AND ACCEPTANCE OF FASTENERS

- Plating Inspection

Most of the platings were discussed earlier in the plating section. Inspection methods for all of them would require too much text, so we will limit our coverage here to zinc and cadmium platings, except for visual and adhesive inspections.

Visual inspections and adhesive inspections are applicable to any type of coating.

The substitution of zinc for cadmium, using a dye to mask the color, is a common way to cheat the customer. This is one of the reasons why zinc and cadmium plating inspection is desired.

INSPECTION AND ACCEPTANCE OF FASTENERS

- Plating Inspection (Cont'd)
  - Types of Plating
    - Zinc Electrodeposited Coatings (per ASTM B633)
    - Cadmium Electrodeposited Coatings (per QQ-P-416)
  - Process Control Inspection
    - A record is maintained of the history of each processing bath showing all additions of chemicals or treatment solutions to the unit and the results of all chemical analyses.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Plating Inspection** (Cont'd)
  - **Lot Sampling Inspection**
    - Sampling of each lot for visual examination and plating thickness tests.
  - **Production Control Tests**
    - Adhesion Test
    - Corrosion Test
    - Hydrogen Embrittlement Test

INSPECTION AND ACCEPTANCE OF FASTENERS

- ## Plating Inspection (Cont'd)

  - **Lot Sampling Inspection**

    - A lot consists of plated articles of the same basis metal composition, class, and type, plated and treated under the same conditions, and submitted for inspection at one time.

      - Selected samples from a lot are examined visually for appearance. The plating should be smooth, adherent, uniform in appearance, free from blisters, pits, nodules, burning, and other defects.

      - Plating thickness of selected samples from a lot are measured non-destructively by electronic test, eddy current, magnetic test, test by beta radiation backscatter principle, or x-ray spectrometry. Microscopic or coulometric destructive measurement procedures may also be used to measure plating thickness.

INSPECTION AND ACCEPTANCE OF FASTENERS

- Plating Inspection (Cont'd)
  - Production Control Tests
    - Four plated or prepared test specimens for each of the required adhesion, corrosion and hydrogen embrittlement tests are sampled from production at scheduled times.
    - Adhesion is determined by scraping the surface or shearing with a sharp edge, knife or razor through the plating to the base metal and examining (at four to ten diameters magnification) for evidence of non-adhesion.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Plating Inspection (Cont'd)**

  - **Production Control Tests (Cont'd)**

    - Corrosion resistance is determined by exposing fasteners to a salt spray for a period of 96 hours. After exposure, the presence of corrosion products visible to the unaided eye at normal reading distance is cause for rejection.

    - Hydrogen embrittlement testing is applied to carbon steel parts which have been heat treated to 144 KSI or above. Fasteners are subjected to a minimum sustained tensile load of 85% tensile ultimate for a minimum of 72 hours. No cracking or fracture of the plated parts is permitted.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Sample Size and Rejection Criteria**

  - The random sampling method of picking test specimens is covered in a number of specifications. Bolts, screws, and studs are covered by ASTM F788. Nut sampling methods are covered in ASTM F812.

  - Another sampling specification is ANSI/ASQC Z1.4, which superseded MIL-STD-105.

  - Another quality assurance acceptance specification for fasteners is ANSI B18.181M.

  - The basis for each of these methods is to randomly pick a small sample (see Table 16) of parts. *ANY* failure of the samples rejects the entire lot.

INSPECTION AND ACCEPTANCE OF FASTENERS

Table 16
Sample Size for Visual Inspection and for Seam Inspection
(From ASTM F788)

LOT SIZE*	SAMPLE SIZE
2 to 15	2
16 to 25	3
26 to 90	5
91 to 150	8
151 to 500	13
501 to 1,200	20
1,201 to 10,000	32
10,001 to 35,000	50
35,001 to 150,000	80

* Lot size is the number of products of the same type, size, and strength grade submitted for inspection one at a time.

Random sample visual examination lot size for the presence of quench cracks, forging cracks, bursts, shear bursts, seams, folds, voids, tool marks, and nicks and gouges is shown in the above table.

If all products are found acceptable during visual inspection, the same visual inspection sample size should be further examined for seams using a technique (magnetic particle, eddy current, liquid penetrant, etc.) capable of detecting the presence of seams. (See Table 17 for sample size.)

INSPECTION AND ACCEPTANCE OF FASTENERS

Table 17
Sample Sizes for Macroscopic Examination of Products With Seam Indications (From ASTM F788)

LOT SIZE*	SAMPLE SIZE
1	1
2 to 8	2
9 to 15	3
16 to 25	5
26 to 50	8
51 to 80	13

* Lot size is the number of products showing seam indication(s) during the seam inspection.

Products with seam indications should be set aside and examined according to the sample size shown in the above table for macroscopic examination for seam indications. Products with the most serious indications should be selected for the sample.

If the macroscopic examination does not establish conclusively the acceptability or rejectability of the lot, the product showing the deepest seam should be further examined microscopically.

INSPECTION AND ACCEPTANCE OF FASTENERS

- Lot Traceability, Comingling and Certifications

There has been a lot of publicity on the effects of the Fastener Quality Act (also known as Public Law 101-592, as amended by P.L. 104-113). Although this law was well-intended, it has been "watered down" by various interest groups to where it has few positive attributes left.

➛ Lot Traceability

The customer can ask for the steel manufacturer's name, the lot number and chemical analysis of the wire from which the fasteners were made. From domestic material suppliers, this information is readily available, although the supplier may add a service charge for providing it.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **Lot Traceability, Comingling and Certifications (Cont'd)**

  ➛ **Comingling (Per Fastener Quality Act)**

  Each fastener manufacturer must register his trademark with ASME. He must also stamp his trademark on all fasteners *COVERED BY THE LAW*. The minimum sizes covered are ¼ inch and 5mm diameters. Nearly all smaller sizes are excluded from the law. If the fasteners have not been exempted from the law by the numerous loopholes, they are now restricted by the comingling rule. This rule limits the comingling of like fasteners (same geometry, material, and strength) to 2 separate lots in the same container.

INSPECTION AND ACCEPTANCE OF FASTENERS

- Sample Size and Rejection Criteria

  ➛ Certifications

  A customer can demand certifications such as material lot number(s), chemical analysis report(s), and tensile test data for his fastener order. This documentation is notarized and is legally binding on the supplier's part. For this reason, some distributors are asking certification charges that far exceed the cost of the fasteners to discourage certification requests.

  If the distributor is not required to provide certifications, he is not legally responsible for his fasteners.

INSPECTION AND ACCEPTANCE OF FASTENERS

- Inspection and Test Standards

There are numerous specifications, many of which have already been mentioned, for test and inspection methods. An attempt was made to list as many as possible in the Appendices.

Additional general references are also given in the Appendices.

Since so many fastener tests are done per MIL-STD-1312, a summary of its contents is given here.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **MIL-STD-1312 Test Methods**

  - MIL-STD-1312 establishes standard methods for testing fasteners in both the International System (SI) of units and the Inch-Pound (I-P) system of units.

  - Standard test methods yield reproducible data and design allowables for use in research, development, procurement, or product application.

  - Each test method is prepared as an individual document in the format of a bookform standard.

  - The individual bookform standard is identified by the basic document identifier MIL-STD-1312 for the I-P system of measurement and DOD-STD-1312 for the SI system of measurement, followed by a sequential dash number to identify the specific test method.

INSPECTION AND ACCEPTANCE OF FASTENERS

- **MIL-STD-1312 Test Methods (Cont'd)**

MIL-STD-1312 dash numbers for methods in the I-P measurement system

-1 Salt Spray
-2 Interaction
-3 Humidity
-4 Lap Joint Shear
-5 Stress Durability
-6 Hardness
-7 Vibration
-8 Tensile Strength
-9 Stress Corrosion
-10 Stress Rupture
-11 Tension Fatigue
-12 Thickness of Metallic Coatings
-13 Double Shear
-14 Stress Durability (Internally Threaded Fasteners)

INSPECTION AND ACCEPTANCE OF FASTENERS

- **MIL-STD-1312 Test Methods** (Cont'd)

MIL-STD-1312 dash numbers for methods in the I-P measurement system (Cont'd)

-15 Torque-Tension

-16 Clamping Force for Installation Formed Fasteners

-17 Stress Relaxation

-18 Elevated Temperature Tensile Strength

-19 Fastener Sealing

-20 Single Shear

-21 Shear Joint Fatigue Constant Amplitude

-22 Receptacle Push-out, Panel Fasteners

-23 Tensile Strength of Panel Fasteners

-24 Receptacle Torque-out, Panel Fasteners

INSPECTION AND ACCEPTANCE OF FASTENERS

- **MIL-STD-1312 Test Methods (Cont'd)**

MIL-STD-1312 dash numbers for methods in the I-P measurement system (Cont'd)

-25 Driving Recess Torque (Quality Conformance Test)

-26 Structural Panel Fastener Lap Joint Shear

-27 Panel Fastener Sheet Pull-up

-28 Elevated Temperature Double Shear

-29 Shank Expanding Characteristics

-30 Sheet Pull-up of Blind Fasteners

-31 Locking Torque

-32 Barrel Nut Tensile Test

INSPECTION AND ACCEPTANCE OF FASTENERS

- **DOD-STD-1312 Test Methods (Metric)**

DOD-STD-1312 dash numbers for methods in the SI measurement system

-105 Stress Durability
-107 Vibration
-108 Tensile Strength
-109 Stress Corrosion
-111 Tension Fatigue
-113 Double Shear

Do's and Dont's for Fastener Design

- Introduction

  A list of common sense guidelines for fastener design should be available for all engineers. The ones given here are not complete, as a new design can always generate a new guideline. However, these guidelines should be used as a designer's checklist.

- Give enough information on the drawing to *FULLY DEFINE* the fasteners you want. This should include material, hardness, plating, and geometry, either by calling out specifications or actual definitions on the drawings.

DO'S AND DONT'S FOR FASTENER DESIGN

- Use a nut that is softer than the bolt.
- No feather edges on sheets in joint.
- Match drill for countersunk holes.
- Use floating nutplates for critical designs.
- Determine environmental conditions before selecting materials or coatings for fasteners.
- Design shear fasteners to be critical in bearing.
- Don't use jam nuts for locking.
- Check alignment of fasteners before final assembly.
- Avoid head bending.

DO'S AND DONT'S FOR FASTENER DESIGN

- Follow fastener edge distance and spacing guidelines.
- Don't use fasteners that look alike but are made of different materials.
- Don't use fine and coarse threads in the same assembly unless there is a large difference in the fastener diameters.
- Don't mix metric and inch fasteners in a design.
- Verify that you have the fasteners that you specified. Demand traceability.
- Use inserts in soft materials to avoid fastener pull-out.
- If the dominant fastener load is shear, don't use a high torque on the fastener. The tension and shear loads must be combined and compared to the total fastener strength.

DO'S AND DONT'S FOR FASTENER DESIGN

- Avoid tapped holes as much as possible.
- Use hardened washers under both the bolt head and nut, if possible.
- Don't torque a fastener above its yield point.
- Use of lubricants lowers the coefficient of friction, so the torque values should be lowered accordingly to avoid bolt overload.
- Torque tables are only guidelines. The design engineer should determine the torque values for his design.
- Fasteners loaded in fatigue should be torqued to near-yield values.

FREQUENTLY ASKED QUESTIONS FOR FASTENER DESIGN

Q1. What torque value should I use?

A1. Look at material strength, lubrication, loading, differential expansion, etc. Give a torque range for the mechanic to work in.

Q2. Should locking torque be added to the regular torque?

A2. First, see if the locking torque is within MSFC-STD-486 guidelines. Then add the locking torque to the regular torque, checking to see that the total torque does not go above fastener yield.

FREQUENTLY ASKED QUESTIONS FOR FASTENER DESIGN

Q3. Do I have to use inserts in aluminum if I install steel bolts?

A3. Try to avoid tapped holes, if possible. If you must have tapped holes, you can delete inserts if galvanic corrosion and disassembly do not need to be considered. There must also be enough depth in the tapped hole to develop bolt strength; otherwise, you need inserts.

Q4. Why did the heads pop off my socket head screws within a few hours after installation?

A4. In spite of all the inspection procedures previously covered, the depth of socket is not measured. If the socket depth goes below the bottom face of the head, the net cross-sectional area is drastically reduced and will probably fail.

FREQUENTLY ASKED QUESTIONS FOR FASTENER DESIGN

Q5. **Assembly/Disassembly life cycles for a locking fastener?**

A5. A deformed thread is usually good for about 10 to 15 assembly/disassembly cycles. A nylon or plastic plug/strip is good for 5 to 10 cycles. However, running (prevailing) torques should be checked against the values given in MSFC-STD-486. If the values are outside the values of MSFC-STD-486, the fasteners should be discarded.

Q6. **Assembly/disassembly life for a fastener?**

A6. If a fastener was torqued above its yield, it *SHOULD NOT* be reused at all. The number of cycles allowed for a fastener torqued below yield is a function of the design criticality. Aircraft fasteners are not supposed to be reused at all.

FREQUENTLY ASKED QUESTIONS FOR FASTENER DESIGN

Q7. What do I use for a shear area when I have a bolt in shear in a tapped hole?

A7. A bolt loaded in shear and installed in a tapped hole is not a good design. However, if you can't avoid this condition, use the root area of the thread for a shear area. If the design is critical include a stress concentration factor of 2.7 to 6.7 (Ref: Peterson) for standard threads.

FREQUENTLY ASKED QUESTIONS FOR FASTENER DESIGN

Q8. Should I use a washer under the head and nut of a bolted assembly?

A8. Hardened washers should be used under both the head and nut to:

- Distribute the contact compressive loads
- Provide a smooth hard surface to minimize the friction forces during torquing
- Provide a counterbored inner washer diameter to avoid point contact on the head fillet.

FREQUENTLY ASKED QUESTIONS FOR FASTENER DESIGN

Q9. When should I use coarse, fine, or extra-fine threads?

A9. Whether you like it or not, fasteners below No. 10 (.190 in.) have coarse threads unless special ordered, due to the difficulty of making very fine threads.

- Coarse threads are easier to install without cross-threading so they are the most popular thread in the corrosive industrial world.
- Fine threads are more easily adjusted and have a larger root diameter than coarse threads. The aerospace world uses fine threads.
- Extra-fine threads are used for fine adjustments such as jacking screws, precision mechanisms, and tapped holes in thin materials.

FREQUENTLY ASKED QUESTIONS FOR FASTENER DESIGN

Q10. What is the difference between a screw and a bolt?

A10. There are many answers for this question, none of which are universally acceptable.

- A screw is installed in a tapped or drilled hole; a bolt is threaded into a nut.
- A bolt is installed with an external wrench; a screw is installed with an internal driver.
- A bolt has a shank diameter equal to the minor diameter of the threads; a screw shank diameter equals the thread major diameter.
- A screw is threaded all the way to the bottom of the head; a bolt has some unthreaded shank.

Note that all of these definitions have overlaps, so you need to clearly specify what you want in geometric terms or a corresponding part number.

FREQUENTLY ASKED QUESTIONS FOR FASTENER DESIGN

Q11. When should I use threaded studs with nuts instead of fasteners in tapped holes?

A11. A threaded stud works better than a fastener in a tapped hole, as it rusts in place and is usually not removed. A nut is easier than a bolt to soak loose and remove. The damaged stud protruding threads can be cleaned and "chased" and a new nut can be installed with much more success than removing a corroded fastener from a tapped hole.